# 10. Piston Machines

Piston Machi

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Piston machines are the most used power and work machines in the mechanical engineering industry. The piston machines are divided in so-called reciprocating and rotary piston machines. With the first one a reciprocating motion is transformed to a rotary motion in the case of the power machine and conversely in the case of the working machine. Today rotary piston machines are almost exclusively used as work machines. Important innovations and intensive researches are practiced particularly for the use of the piston machines as an internal combustion engine. Therefore the mixture formation and the combustion process, with their consequences in terms of emission and fuel-consumption are int the center of attention.



## 10.1 Foundations of Piston Machines

### 10.1.1 Definitions

Piston machines employ a moving displacer (also called a piston) to convert a medium's potential energy into kinetic energy or vice versa, i. e., they use the movement of the displacer to increase the energy content of the medium. This occurs in a working chamber that can be altered by the displacer motion.

In piston machines, the moving displacer effects both the charge cycle (filling and draining of the medium) and the work cycle (expansion and compression). The characteristic mode of operation for piston machines is a self-contained working chamber that varies periodically due the piston's movement.

Piston machines can be classified according to their method of operation, the piston motion and the medium used (Fig. 10.1). When piston machines are classified according to their method of operation then power and work machines are differentiated.

A power machine converts a medium's potential energy into mechanical energy. Power machines include engines (pneumatic engine, hydraulic motor) and thermal machines (steam engines, combustion engines). Work machines on the other hand utilize mechanical energy to increase the energy of the medium being

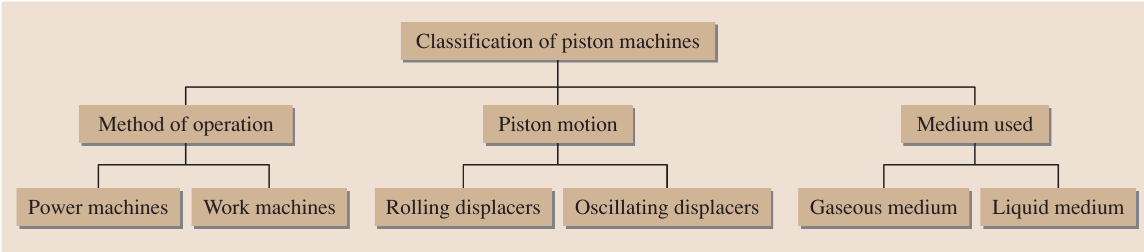



**Fig. 10.1** Classification of piston machines (after [10.1])

conveyed. Work machines include compressors and pumps.

Rotating and oscillating displacer motions are other potential classification.

In addition, piston machines can be classified according to the medium used, which can be gaseous or liquid.

### Reciprocating Machines

When a piston machine's oscillating displacer executes a linear motion, it is called a reciprocating machine (Fig. 10.2). A cylindrical displacer that moves between two end positions, top and bottom dead center (TDC and BDC), is characteristic. The working chamber is calculated from the piston diameter and the distance between the two dead centers, the so-called stroke $s$ (see *Working Chamber* below), and is therefore also referred to as the displacement.

### Rotary Piston Machines

Machines of this type are characterized by a rotating displacer. The medium can flow axially or radially to the piston axis. An axial direction of flow is found, for instance, in screw-type compressors, screw pumps or eccentric screw pumps. Rotary piston compressors, Roots blowers and gear pumps are examples in which the medium flows radially to the piston axis. In Wankel engines (Fig. 10.3), however, the medium flows axially as well as radially, depending on the design.

### Cylinder Configuration

Differently configuring the interrelation of the cylinders makes it possible to produce various types of piston machines.

The inline and V configuration in particular and the boxer variant to a lesser degree are choices for combustion engines. The V design is compact and produces high output per unit volume. Its compact design makes manufacturing it more complex and expensive though. Moreover, accessibility is impaired, as a result of which

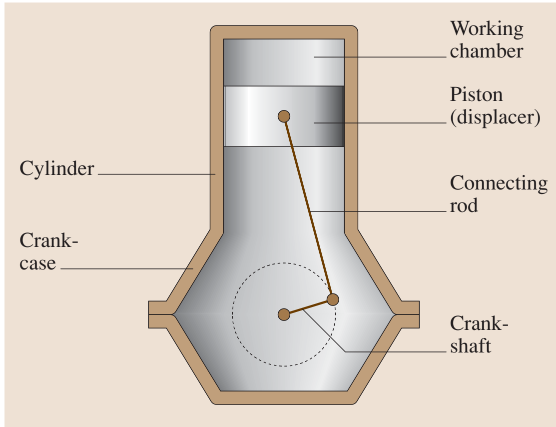



**Fig. 10.2** Reciprocating machine

the time and effort required for maintenance increase along with maintenance costs. The advantages of the boxer engine are its overall length and height, though at the expense of width. The W configuration is frequently employed for piston compressors, yet relatively infrequently for combustion engines. The radial configuration is considered obsolete. It was used early on for

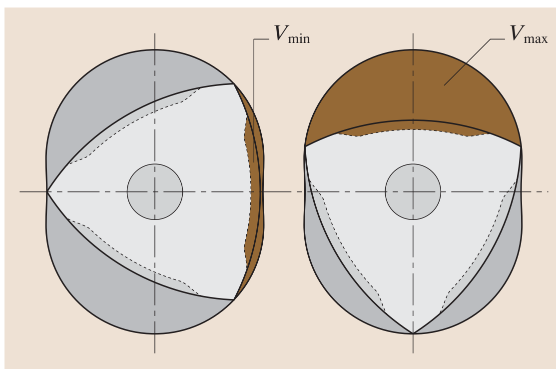



**Fig. 10.3** Wankel engine as an example of a rotary piston machine (after [10.2])

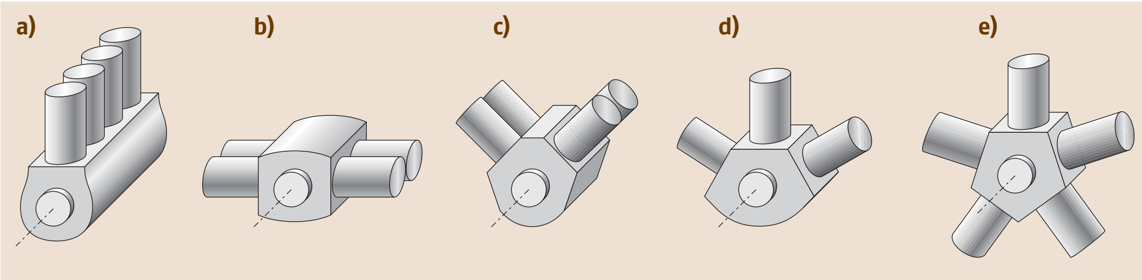



**Fig. 10.4a–e** Cylinder configurations (after [10.1]) **(a)** in-line machine **(b)** opposed-cylinder machine **(c)** V machine **(d)** W machine **(e)** radial machine

aircraft engines because of the good air cooling resulting from the design and the minimal axial extension.

A multi-cylinder design helps increase machine performance for combustion engines. In compressors, multistage compression is achieved by additionally using varying piston diameters. Increasing the number of cylinders boosts running smoothness. A larger number of cylinders adversely affects production and maintenance costs. Moreover, a more-complex design increases susceptibility to failure. Cylinder configuration, unit size, and the stiffness of the crankshaft can limit the number of cylinders.

### Working Chamber

The displacer's movement causes the actual working chamber volume $V_a$ to vary between the volume limits $V_{min}$ and $V_{max}$. $V_{min}$ can be a design-related clearance volume $V_S$ or, in the case of combustion engines, a compression volume $V_c$ contingent on the working process. Thus $V_{min} \leq V_a \leq V_{max}$ applies.

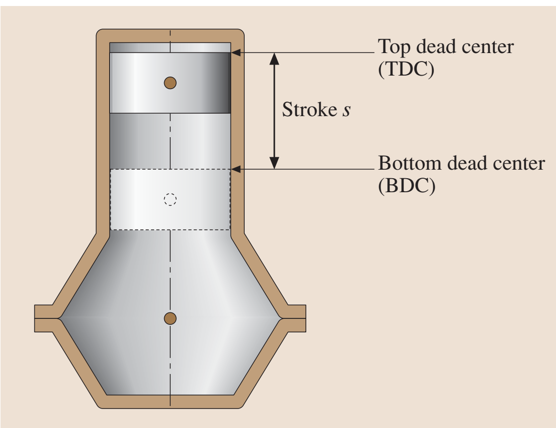



**Fig. 10.5** The stroke $s$ corresponds to the distance between top and bottom dead center

In a reciprocating machine, the maximum piston displacement $V_A$ or swept volume $V_h$ corresponds to the volume resulting from the product of the piston surface $A_p$ and the stroke $s$ (10.1). The stroke corresponds to the distance that the piston covers between the top and bottom dead centers (Fig. 10.5)

$$V_A = V_h = A_p s = s \cdot \pi \cdot D^2/4 . \tag{10.1}$$

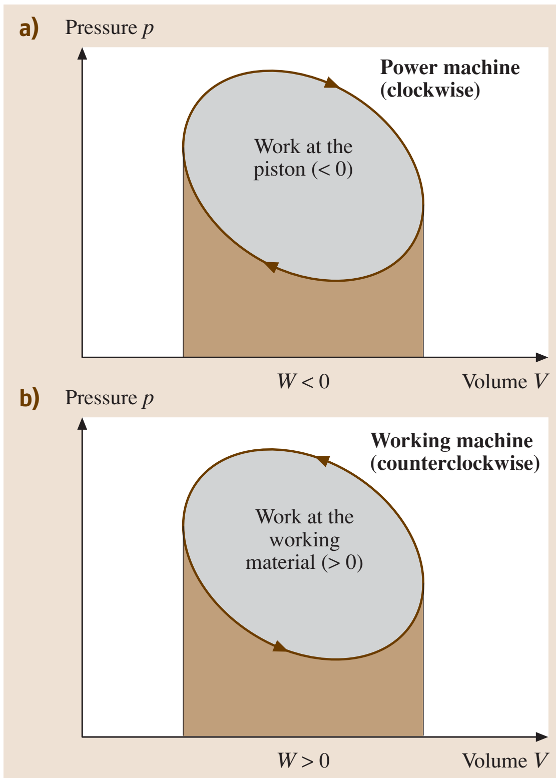



**Fig. 10.6** $p$–$V$ diagrams of power and work machines (after [10.3])

Defining a cylinder's volume (corresponding to the maximum volume) requires the incorporation of the minimum volume

$$V_{\text{cyl}} = V_{\max} = V_{\text{h}} + V_{\min} \,. \tag{10.2}$$

For machines with a multi-cylinder design, the total piston swept volume $V_{\text{H}}$ follows from the number of cylinders $z$ and the swept piston volume $V_{\text{h}}$

$$V_{\text{H}} = zV_{\text{h}} = zA_{\text{p}}s \,. \tag{10.3}$$

Using (10.2), the total working chamber of a combustion engine is calculated as

$$V_{\text{working-chamber}} = (V_{\text{h}} + V_{\min}) \,. \tag{10.4}$$

## 10.1.2 Ideal and Real Piston Machines

### The Ideal Combustion Cycle

A cycle is a succession of a material's changes of state until it returns to its initial state. The cycle serves as the foundation for evaluating the thermodynamics of processes.

The $p$–$V$ diagram is used to represent the development of cycles (Fig. 10.6). Its enclosed area corresponds to the cycle work $W$.

Figure 10.6 is a general representation of the development of the cycles of a power and a work machine. The different direction of rotation of each process and the resultant sign of the cycle work are characteristic here. Since the work done by the machine's piston can be utilized (effective work), it is defined negatively, while the work expended on the working material is defined positively. The cycle work is calculated from the total of both values and corresponds to the area within the circle

$$W = W_{\text{at-the-piston}} + W_{\text{at-the-working-material}} \,, \tag{10.5}$$

$$W = \oint V \mathrm{d}p = -\oint p \mathrm{d}V \,. \tag{10.6}$$

A clockwise cycle profile with negative cycle work results for the power machine, while a counterclockwise cycle profile with positive cycle work results for the work machine.

Three examples of cycles based on the Carnot process (Chap. 4) are described below and are referred to as reference cycles. They are subject to the following premises:

1. State changes are infinitely slow.
2. The working chamber is adiabatically and hermetically sealed.
3. Fluid is exchanged without any change in the state variables (mass, pressure, temperature).

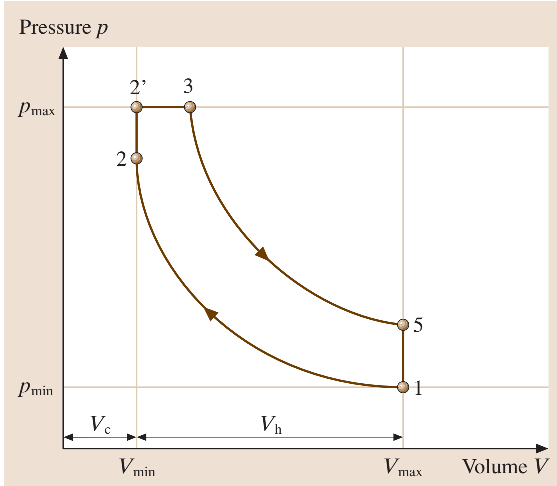



**Fig. 10.7** $p$–$V$ diagram of a combustion engine (Seiliger process) (after [10.4])

*Combustion Engines.* The Seiliger process constitutes a reference process for diesel engines. Adiabatic compression causes the pressure to rise until it is slightly below the maximum cylinder pressure. The internal combustion of a fuel mass is followed by an isochoric and isobaric input of heat, as a result of which the maximum cylinder pressure is reached. This is followed by an isentropic expansion up to the initial volume. The

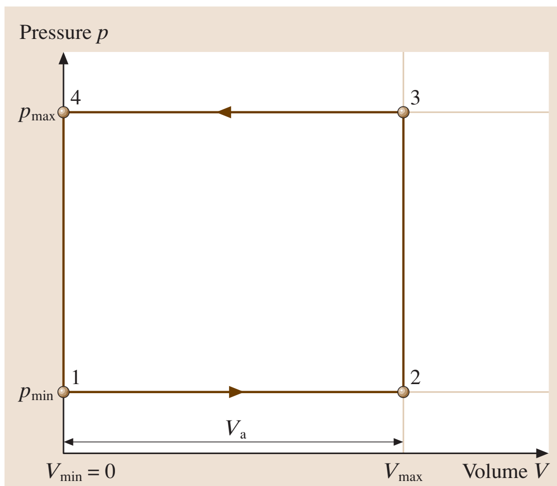



**Fig. 10.8** $p$–$V$ diagram of a positive displacement pump (after [10.4])

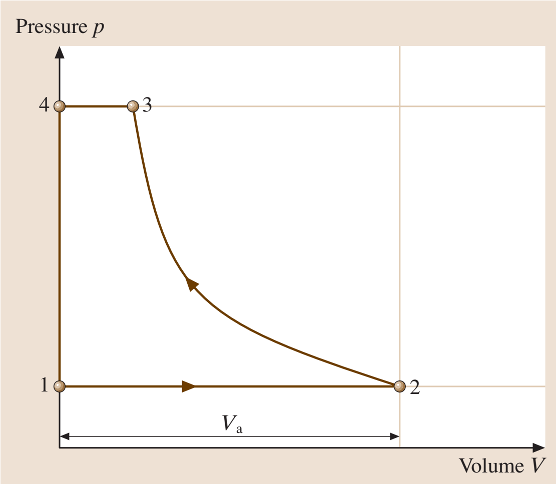



**Fig. 10.9** $p$–$V$ diagram of a positive displacement compressor (after [10.4])

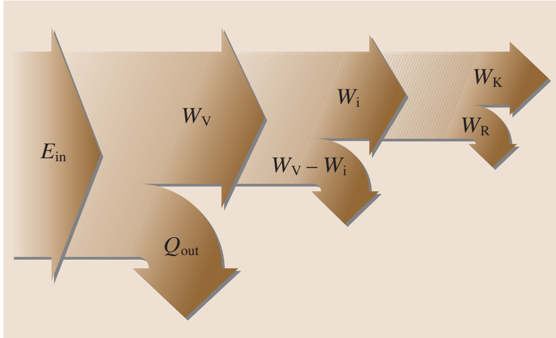



**Fig. 10.10** Schematic diagram of a real power machine's losses (combustion engine)

initial state has been reached again when the heat extraction is isochoric.

*Positive Displacement Pump.* In an initial step, the working chamber is filled with an incompressible fluid. When the maximum volume $V_{max}$ is reached, the pressure climbs isochorically to the maximum pressure $p_{max}$. The fluid is expelled at the maximum pressure. When the working chamber has been completely discharged, the pressure falls isochorically, returning to the initial state.

*Positive Displacement Compressor.* Analogous to the positive displacement pump, the working chamber is filled isobarically with a fluid up to the volume limit. When the positive displacement compressor's cycle is ideal, the low compression work causes the pressure to rise isothermally. When the maximum pressure has been reached, the fluid becomes isobaric and is expelled isochorically until it has been completely discharged.

### Real Machines

While lossless state changes are assumed for an ideal combustion cycle, losses in the real machine occur as irreversible subcycles when the state changes. Infinitely slow (quasistatic) state changes would be needed to prevent irreversible cycles.

Deviations from the ideal cycle occurring in real machines are caused by:

1. Wall heat losses, i. e., heat exchange with the system boundaries (e.g., with the working chamber's walls and the piston)
2. Irreversibilities when states change; changing material values during the cycle
3. Losses when energy is transferred (e.g., friction in the machine, flow losses)
4. Leaks because the working chamber is not absolutely sealed

### Efficiency

The aforementioned deviations are expressed by the efficiency, which represents the relationship between utility and effort. For heat engines, the total effort corresponds to the total quantity of input heat, while the utility is the outcome of an energy conversion process.

*Power Machine.* Figure 10.10 is a schematic diagram of a real power machine's losses.

Heat exchange with the system boundaries represents the greatest percentage loss of input energy. The relationship between the cycle energy of an ideal power machine's reference cycle and the input energy is expressed with the efficiency of the reference cycle and referred to as the efficiency of the ideal machine

$$\eta_v = W_v / E_{in} \; . \tag{10.7}$$

The irreversibilities of a cycle's state changes cause further losses. The related efficiency specifies the indicated cycle's approximation of the reference cycle and is referred to as a cycle's efficiency

$$\eta_g = W_i / W_v \; . \tag{10.8}$$

The work against friction $W_R$ subsumes the losses occurring during the transfer of energy. The relationship between the effective work or the useful work that can be extracted at the clutch $W_K$ and the indicated work is referred to as the mechanical efficiency

$$\eta_m = W_K / W_i \; . \tag{10.9}$$

Consequently, a real power machine's coupling efficiency is calculated from the efficiency chain

$$\eta_K = \eta_v \eta_g \eta_m = (W_v/E_{in})(W_i/W_v)(W_K/W_i) = W_K/E_{in} . \quad (10.10)$$

This efficiency represents all losses occurring in the power machine, which leads to the output useful work (clutch work) $W_K$.

*Work Machine.* Analogous to the power machine, Fig. 10.11 is a schematic diagram of a real work machine's losses.

The following relationships likewise apply to the mechanical efficiency

$$\eta_m = W_i/W_K , \quad (10.11)$$

internal efficiency

$$\eta_g = W_V/W_i \quad (10.12)$$

and system efficiency

$$\eta_A = E_{benefit}/W_V . \quad (10.13)$$

Consequently, a work machine's total efficiency is calculated from the chain of efficiencies

$$\eta = \eta_m \eta_g \eta_A = (W_i/W_K)(W_V/W_i)(E_{benefit}/W_V) = E_{benefit}/W_K . \quad (10.14)$$

### Specific Energy

The relationship between the piston machine's output energy $W_e$ and the corresponding piston swept volume is referred to as the volume-specific energy $w_e$

$$w_e = W_e/V_H . \quad (10.15)$$

In combustion engines, the unit of the volume-specific energy $w_e$ is denoted as $kJ/dm^3$. Work machines' mass-specific energy $w'$ is obtained from the relationship of the requisite work energy $W_a$ and the mass of the fluid delivered per working cycle $m_f$. It is specified, for instance, in kJ/kg

$$w' = W_a/m_f . \quad (10.16)$$

The more losses that occur in the work machine, the more energy is needed to deliver the same amount of fluid. Consequently, the mass-specific work also increases as losses occur.

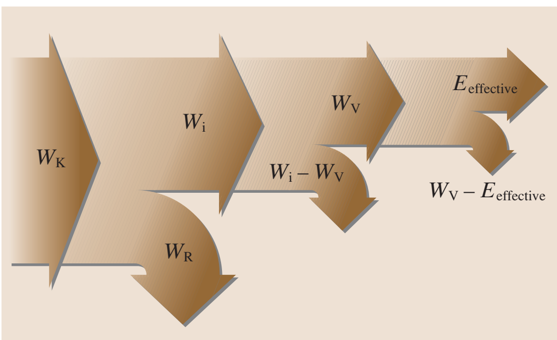



**Fig. 10.11** Schematic diagram of a real work machine's losses

### Power

A machine's power corresponds to the energy output (power machine) or input (work machine) per unit time.

$$P = \frac{W}{t} . \quad (10.17)$$

Moreover, the power can be calculated using the product of the torque $M_d$ and angular velocity $\omega$

$$P = M_d\omega = M_d 2\pi n , \quad (10.18)$$

where $\omega$ is the angular velocity and $n$ is the engine speed or with the aid of the cycle energy $W_K$

$$P = \frac{W_K}{t_{ASP}} = W_K f_{ASP} , \quad (10.19)$$

where $t_{ASP} = i/2n$ is the time for one working cycle and $f_{ASP} = 1/t_{ASP}$ is the working cycle frequency; $i$ is the number of cycles or piston strokes per working cycle. Expressing this using the mass flow $\dot{m}$ and the mass delivered per working cycle $m_{ASP}$ leads to

$$P = \frac{W_K}{m_{ASP}}\dot{m} . \quad (10.20)$$

## 10.1.3 Reciprocating Machines

### Types of Transmissions

Different types of transmissions transmit energy between piston and crankshaft, e.g., crankshaft drive, swash plate drive, cam drive and eccentric drive. The simple engineering and dimensioning of a reciprocating machine's important parameters (e.g., stroke and compression) are the reason why a crankshaft drive is frequently used as the transmission.

The connecting rod transmits energy between the piston and crankshaft (Fig. 10.2). In the process, the piston's oscillating motion is converted into the crankshaft's rotating motion in order to improve the further transmission of energy. Accordingly, the motion is transformed conversely when energy is transmitted from the crankshaft to the piston. The crankshaft drive can be constructed as a plunger piston or crosshead version. The crosshead serves to relieve the piston from

the lateral piston force, particularly in large piston machines.

### Kinematics

The piston executes a motion characterized by accelerations and decelerations between the two dead centers. The piston's linear, irregular motion at the crankshaft is converted by the connecting rod into a rotating motion with constant angular velocity. The instantaneous distance covered by the piston, related to the top dead center, is designated the piston travel $x$ (Fig. 10.12). The maximum piston travel, i.e., the distance between the two dead centers, is defined as the stroke $s$. This corresponds to twice the distance between crankshaft journal and shaft journal (crank radius $r$).

Piston travel can be specified as a function of the piston angle $\alpha$

$$x = l + r - (l\cos\beta + r\cos\alpha)\,. \tag{10.21}$$

By substituting the connection rod angular travel $\beta$ with $\alpha$ by using the relation

$$l\sin\beta = r\sin\alpha \tag{10.22}$$

and the connecting rod ratio

$$\lambda = r/l\,, \tag{10.23}$$

the formula for piston travel is obtained as

$$x = r(1-\cos\alpha) + l\left(1-\sqrt{1-\lambda^2\sin^2\alpha}\right)\,, \tag{10.24}$$

where $\sin\beta = \lambda\sin\alpha$ and $\cos\beta = \sqrt{1-\lambda^2\sin^2\alpha}$.

Expanding the expression under the root into a Taylor polynomial produces a complex equation. In practice, only the first two orders of the Taylor polynomial are applied since they already yield sufficiently precise results for the piston travel $x$

$$x \approx r\left(1-\cos\alpha + \frac{\lambda}{2}\sin^2\alpha\right)\,. \tag{10.25}$$

Piston speed can be stated as an average or instantaneous speed.

The average piston speed is calculated based on the simple relation between distance covered and the necessary time expended. Twice the piston stroke is covered during one revolution. This yields the formula for the average piston speed

$$c_\mathrm{m} = \frac{2s}{\frac{1}{n}} = 2sn\,. \tag{10.26}$$

The following numerical value equation is also frequently applied

$$c_\mathrm{m} = \frac{sn}{30}\,. \tag{10.27}$$

The unit of revolution has to be specified in 1/min or $\mathrm{min}^{-1}$ and rpm respectively in order to obtain the average piston speed in m/s.

If the actual piston speed $v$ is required, piston travel must be differentiated with respect to the time $t$, remembering that piston travel is only a function of the piston angle

$$v = \frac{\mathrm{d}s}{\mathrm{d}t} = \frac{\mathrm{d}s}{\mathrm{d}\alpha}\frac{\mathrm{d}\alpha}{\mathrm{d}t}\,, \tag{10.28}$$

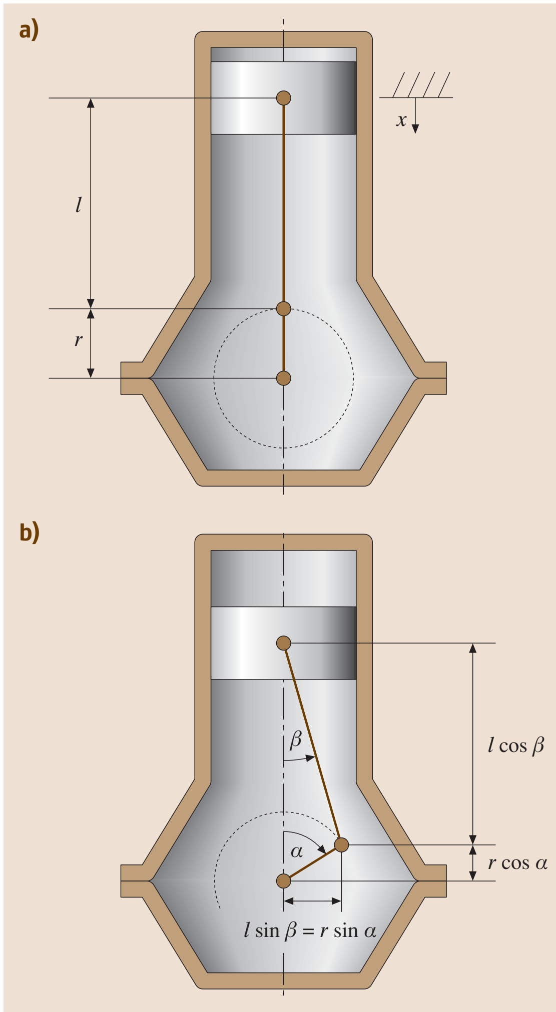



**Fig. 10.12** Piston travel $x$ and crankshaft drive geometry

where $d\alpha/dt$ is the crankshaft angular velocity $\omega$.

Differentiating (10.25) yields an approximate equation for the piston speed:

$$v \approx \omega r \left[ \sin\alpha + \frac{\lambda}{2}\sin(2\alpha) \right] . \tag{10.29}$$

Analogous to the piston speed, an approximate value can be derived for the piston acceleration

$$a \approx \omega^2 r [\cos\alpha + \lambda \cos(2\alpha)] . \tag{10.30}$$

Equations (10.25), (10.29) and (10.30) do not yield any precise results when speeds are high (e.g., racing engines). The exact piston travel and the corresponding derivatives for speed and acceleration would have to be used for this application.

### Forces

*Fluid Forces.* The state change of the fluid enclosed in the working chamber induced by the oscillating motion of the displacer causes a force that is dependent on the piston surface $A_K$ and the fluid pressure $p_F$ exerted on the piston. Allowing for the atmospheric pressure $p_0$, the fluid force $F_F$ of a single-action piston is calculated with the following equation

$$F_F = A_K(p_F - p_0) . \tag{10.31}$$

An equal force acts on the cylinder cover. The bolted connections, e.g., between the cylinder cover and the housing, create a closed linkage so that the forces in the machine balance (Fig. 10.13).

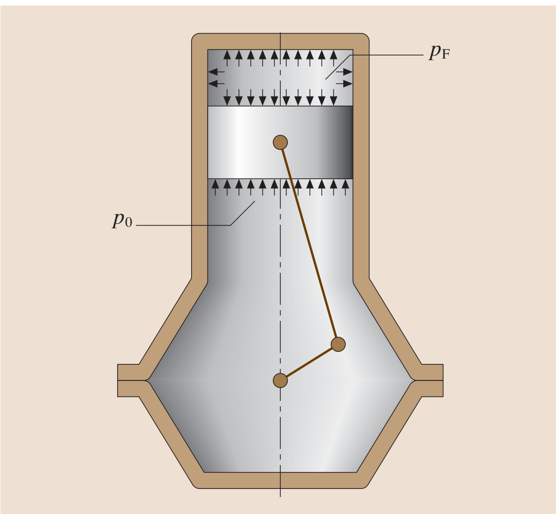



**Fig. 10.13** Pressures on the piston

*Inertial Forces.* The irregular motions of the translationally moving structural parts cause periodic inertial forces to arise, which act as oscillators. The centripetal acceleration generates rotary inertial forces on the connecting rod and crankshaft suspensions. Therefore they have to be balanced. This can be easily and completely done for rotary piston machines, but only partially or with considerable effort for reciprocating machines. The inertial forces are classified as rotating and oscillating based on the different types of motion of the transmission components.

The rotating inertial forces $F_{mr}$ are calculated with the formula

$$F_{mr} = m_r r \omega^2 . \tag{10.32}$$

The rotating masses $m_r$ are concentrated in the crankshaft journal at the distance $r$ from the center of rotation and consist of the masses of the crankshaft journal $m_z$, the crankshaft web (converted to the crank radius), and the rotating portion of the connecting rod (10.36).

The oscillating and rotating motion of the connecting rod cause the mass to be distributed to two points

$$m_{Pl} = m_{Plo} + m_{Plr} , \tag{10.33}$$

where $m_{Pl}$ is the connecting rod mass, $m_{Plo}$ is the oscillating mass fraction, $m_{Plr}$ is the rotating mass fraction, $a$ and $b$ are the distances to the connecting rod center, and $l$ is the distance between the two connecting rod eyes.

The mass $m_{Plo}$ is located in the piston pin and is only involved in the oscillating motion, while $m_{Plr}$ on the crankshaft journal executes a purely rotary motion. Since the distances from the center and the total moment of inertia are maintained, the masses are calculated based on

$$m_{Plo} = \frac{m_{Pl} a}{l} \tag{10.34a}$$

and

$$m_{Plr} = \frac{m_{Pl} b}{l} . \tag{10.34b}$$

As a result, a substitute connecting rod is obtained, the dynamic behavior of which is approximately comparable with a real connecting rod.

Since the center of the web masses $m_W$ does not lie at the center of the crankshaft axis, the masses must be converted to the crank radius. Using the distance $x$ between the center of the web masses and the crankshaft axis as well as the crank radius, the reduced mass $m_{WE}$ is calculated as

$$m_{WE} = m_W \frac{x}{r} . \tag{10.35}$$

Taking the preceding observations and allowing for two web masses, the following result ensues for the rotating inertial forces of the reciprocating machine

$$F_{mr} = (m_Z + m_{Plr} + 2m_{WE})r\omega^2 . \quad (10.36)$$

The irregular piston motion generates oscillating inertial forces $F_{mo}$ as a function of the piston acceleration $a$

$$F_{mo} = m_o a , \quad (10.37)$$

where $m_o = m_K + m_{Plo}$ are the oscillating masses.

Taking the piston acceleration (10.30), the following ensues for the oscillating inertial force

$$F_{mo} \approx m_o\omega^2 r[\cos\alpha + \lambda\cos(2\alpha)] . \quad (10.38)$$

It is divided into oscillating inertial forces of first and second order (I and II)

$$F_{moI} = m_o\omega^2 r\cos\alpha = F_I\cos\alpha , \quad (10.39a)$$

$$F_{moII} = m_o\omega^2 r\lambda\cos(2\alpha) = F_{II}\cos(2\alpha) . \quad (10.39b)$$

The frequency of the first-order forces of inertia corresponds to the crankshaft speed, while the second-order forces of inertia change as the crankshaft speed doubles.

The influence of the stroke/connecting rod ratio $\lambda$ (10.23) makes the amplitude of the second-order forces of inertia less than that of the first order. The extreme values for $F_{moI}$ are in the top and bottom dead center, i.e., at $\alpha = 0°$ and $\alpha = 180°$. Since the frequency of $F_{moII}$ is doubled, the maximum amplitudes are at $\alpha = 0, 90, 180$ and $270°$. This yields the following for the oscillating inertial forces at top dead center

$$F_{mo} = F_{moI} + F_{moII} \quad (10.40)$$

and the following at bottom dead center

$$F_{mo} = F_{moI} - F_{moII} . \quad (10.41)$$

Since there is no closed linkage, the oscillating inertial forces cause the crankcase to oscillate and thus exerts a load on the external engine suspension.

*Forces on the Transmission.* The piston force $F_K$ consists of the fluid force $F_F$ and the oscillating inertial force $F_{os}$. It is broken down into a lateral force $F_N$ and a connecting rod force $F_S$ as follows (Fig. 10.15)

$$F_N = F_K\tan\beta , \quad (10.42a)$$

$$F_S = \frac{F_K}{\cos\beta} . \quad (10.42b)$$

The cylinder wall absorbs the normal force. The connecting rod force divides at the crankshaft journal into a radial and a tangential component ($F_R$ and $F_T$)

$$F_R = F_S\cos(\alpha+\beta) = F_K\frac{\cos(\alpha+\beta)}{\cos\beta} , \quad (10.43a)$$

$$F_T = F_S\sin(\alpha+\beta) = F_K\frac{\sin(\alpha+\beta)}{\cos\beta} . \quad (10.43b)$$

The tangential force and the crank radius produce the torque

$$M_d = F_T r . \quad (10.44)$$

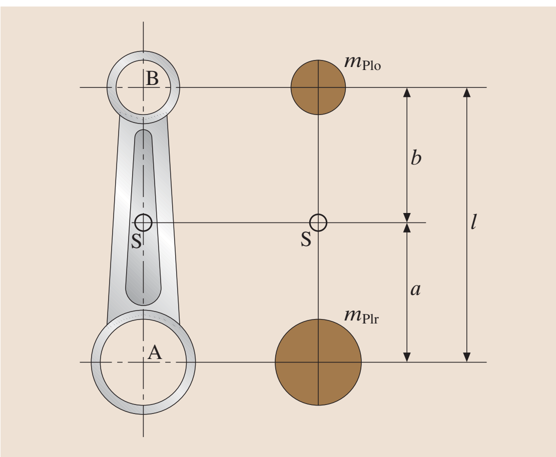



**Fig. 10.14** Distribution of the connecting rod mass (after [10.5])

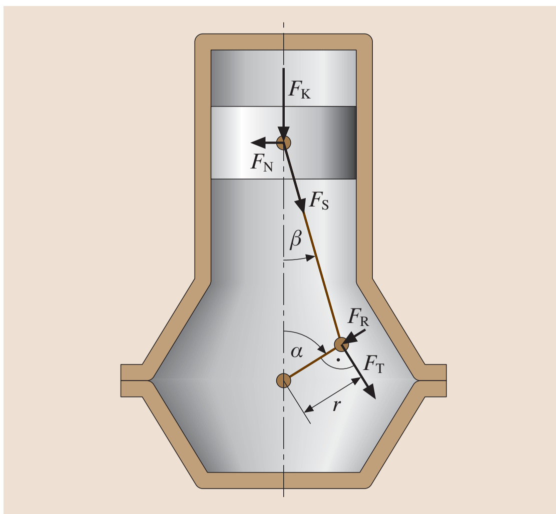



**Fig. 10.15** Forces on the transmission

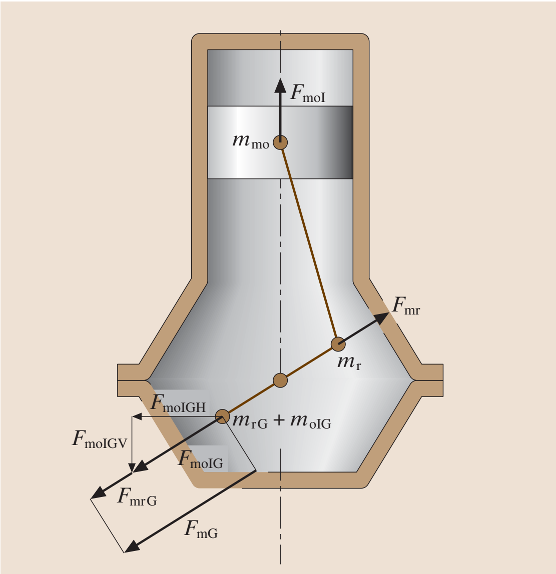



**Fig. 10.16** Counterweight and forces

The direction of the force $F_T$ corresponds to the direction of rotation. The normal force generates the corresponding reaction torque, which acts as a tilting moment on the housing and must be absorbed by the housing suspension.

### Mass Balancing

The periodically varying inertial forces and moments of the transmission's moving points can cause oscillations, which have an effect on the engine suspension and substructures. This load can be counteracted by balancing masses at the crankshaft webs. The configuration of this mass balancing depends on the number and arrangement of the cylinders as well as the distribution of the crank throw.

*Inertial Force Balancing on Single-Cylinder Machines.* Only rotating and oscillating inertial forces, but no moments of inertia, occur on a single-cylinder machine since the plane of symmetry lies in the cylinder axis.

The rotating inertial forces can be balanced relatively easily and completely by two counterweights $m_{rG}$ on the crankshaft (Fig. 10.16). These are offset by 180° with respect to the crankshaft journal and are calculated from the rotating mass $m_r$ and the corresponding distances from the center of rotation as

$$m_{rG} = 0.5 m_r \frac{r}{r_G} , \qquad (10.45)$$

where $r$ is the crank radius and $r_G$ is the distance of the counterweights from the center of rotation.

The distribution of the balancing masses to individual counterweights with the mass $m_{rG}$ is incorporated to prevent an additional moment.

To balance the oscillating inertial forces, (10.38) is applied and broken down into first- and second-order forces of inertia. The change of the first-order forces of inertia corresponds to the frequency of the crankshaft speed and consequently can be partially balanced by counterweights on the crankshaft.

Second-order forces of inertia change twice as fast. Therefore they cannot be balanced by counterweights on the crankshaft.

The oscillating inertial forces act in the direction of the cylinder axis. Hence, only the perpendicular force components of the rotating counterweights can be used as mass balancing. The horizontal force components also generated represent an interfering force. The following equation is used to calculate the counterweights

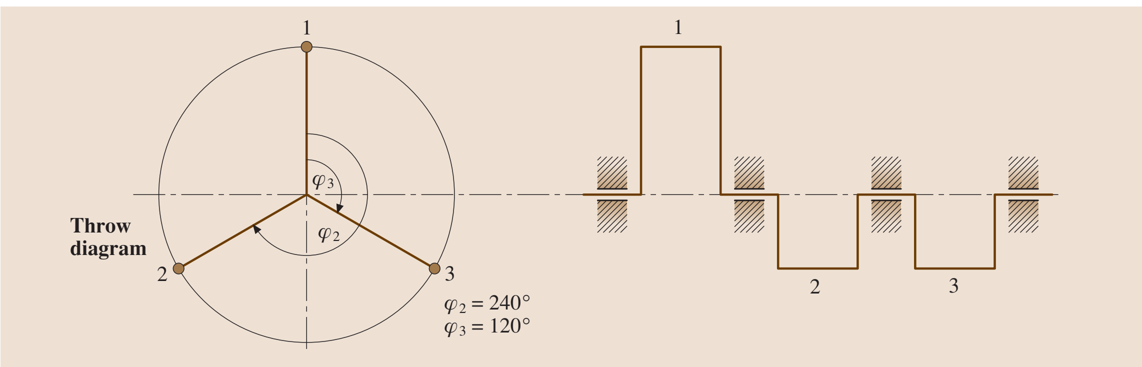



**Fig. 10.17** Cross and longitudinal sections of a three-cylinder crankshaft (after [10.3])

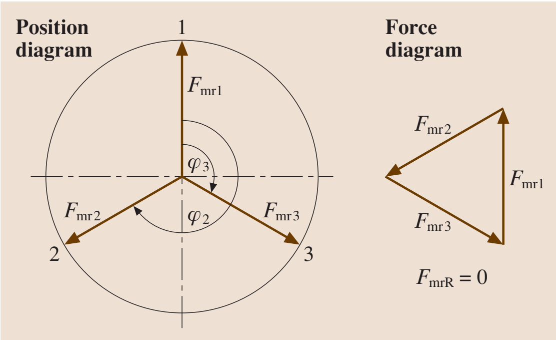



**Fig. 10.18** Determination of the resultants from the rotating inertial forces (after [10.3])

$m_{oIG}$ per web to balance the first-order forces of inertia

$$m_{oIG} = 0.5 m_o \varphi \frac{r}{r_G} , \qquad (10.46)$$

where $\varphi$ represents the proportion of the first-order forces of inertia being balanced. This prevents the influence of the interfering horizontal force components from becoming too great.

*Mass Balancing on Multi-Cylinder Machines.* To balance masses for multi-cylinder engines, the inertial forces are calculated for every cylinder and consolidated in a resultant. The aim of mass balancing is to compensate the inertial forces reciprocally and to keep the resultants as small as possible.

Furthermore, the concentrated loads not engaging in the center of mass cause additional moments of inertia. These are also consolidated into resultants.

The radial and axial arrangement of the crank throw strongly influence the value of the resultants. When designing the crankshaft, attention has to be paid to balancing the concentrated loads. One simple way to do this is by projecting the crankshaft radially and axially. Cross and longitudinal sections of the crankshaft are drawn (Fig. 10.17). The cross section (the left-hand image in Fig. 10.17) is called a throw diagram because of its shape. Along the longitudinal section (the right-hand image in Fig. 10.17) of the crankshaft, the crank throws are numbered from left to right and subsequently transferred to the throw diagram.

*Resultants from the Rotating Inertial Forces.* The rotating inertial forces calculated with (10.32) are transferred in parallel to a common cross-sectional plane. By adding the vectors, the individual forces are consolidated into a resultant $F_{mrR}$ (Fig. 10.18).

The resultant rotates with the crankshaft speed and has a constant value. Hence it does not have to be determined anew for other crankshaft positions, but only rotated by the corresponding angle (Fig. 10.18).

*Resultants from the First-Order Forces of Inertia.* The specific instantaneous value in the cylinder axis is calculated by projection onto these and the total first-order inertial forces in the direction of the cylinder axis is determined by adding their vectors. The maximum value of the first-order forces of inertia is calculated with

$$F_{moI\_max} = m_o \omega^2 r . \qquad (10.47)$$

A simpler approach is often used in practice. The maximum values are likewise applied in the direction of the crank arm. Adding these forces yields a resultant of the maximum values $F_{moI\_max\,R}$ that is projected onto the cylinder axis. The force calculated in this way is the resultant of the first-order forces of inertia $F_{moIR}$ (Fig. 10.19).

The resultant of the maximum values rotates together with the crankshaft as it rotates. In the corresponding crank position, it must be projected

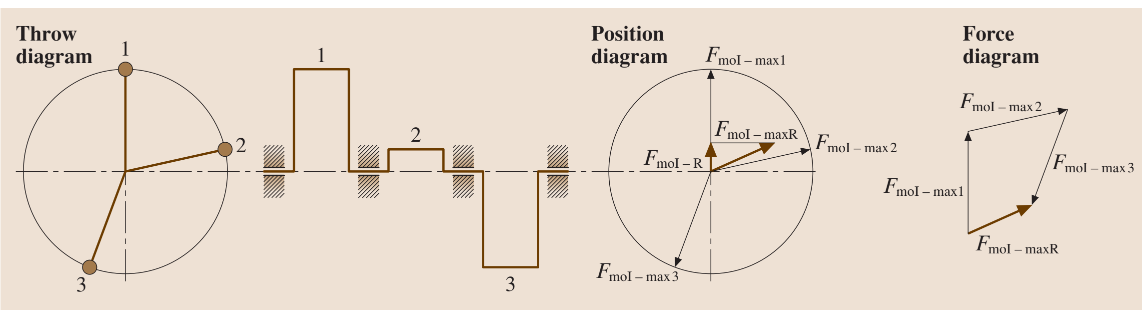



**Fig. 10.19** Determination of the resultants from the first-order forces of inertia (after [10.6])

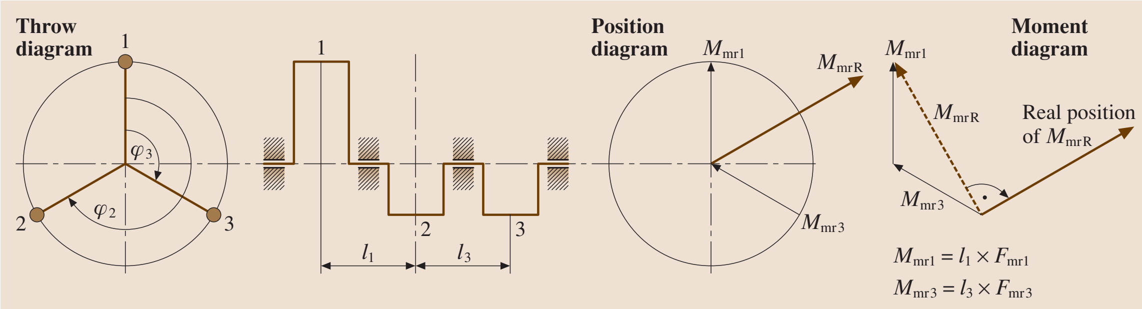



**Fig. 10.20** Determination of the resultant moment of inertia from the rotating inertial forces (after [10.6])

anew onto the cylinder axis in order to obtain the corresponding resultant of the first-order forces of inertia.

*Resultants from the Second-Order Forces of Inertia.* The resultant of the second-order forces is calculated in the same way as the resultant from the first-order forces of inertia. However, the second-order forces of inertia change their value as the crankshaft's angle of rotation doubles. Therefore they cannot be applied in the direction of the crank arms. Rather, they have to be drawn at twice the angle. A force plan is again used to calculate the resultant of the maximum values, which is subsequently transferred to the position diagram. Projection onto the cylinder axis yields the resultant of the second-order forces of inertia.

The rotation of the resultant of the maximum values has to be entered on the site plan at twice the crankshaft angle. The resultant sought is calculated by projection onto the cylinder axis.

*Resultant Moment of Inertia.* The inertial forces in multi-cylinder engines act at a specific distance to the machine's center of gravity and cause moments of inertia. Since their precise determination is extremely involved, a simplified assumption places the center of gravity in the longitudinal section in the crankshaft axis (e.g., the second cylinder's axis in a three-cylinder machine; see the longitudinal crankshaft section in Fig. 10.20). The resulting error is negligible in most cases.

The moment vectors are applied corresponding to the crank arms taken from the throw diagram in the site plan. The direction of the vectors is determined as follows. The vectors in the longitudinal section to the left of the reference point point outward, while the vectors to the right of the reference point point toward the center of the throw diagram.

The value of the resultant moment of inertia is obtained by adding the vectors in the moment diagram. The vector is given its correct position by rotating it clockwise by 90°.

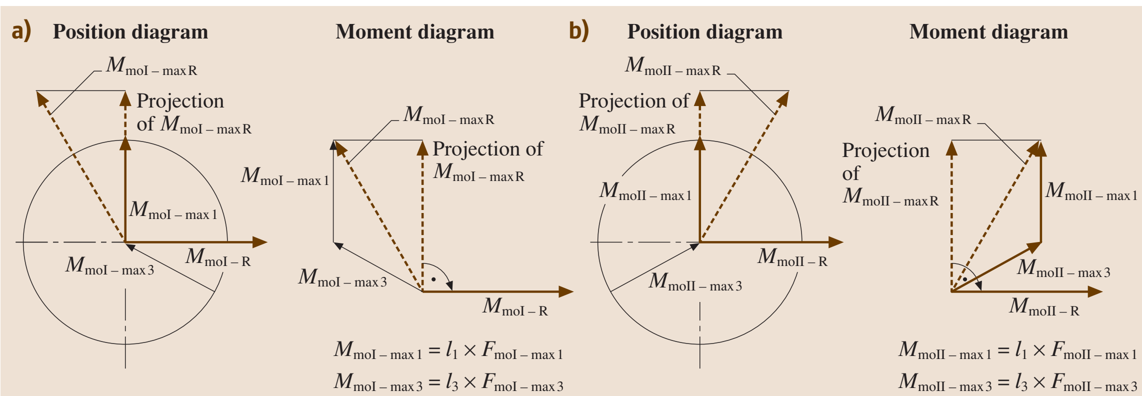



**Fig. 10.21a,b** Determination of the resultant moment of inertia from the oscillating **(a)** first- and **(b)** second-order inertial forces (after [10.6])

A similar approach is employed to determine the resultant first-order moment of inertia. However, the resultant of the maximum values of the moments is projected onto the cylinder axis and then rotated clockwise by 90° (left-hand position and moment diagram in Fig. 10.21).

When the resultant second-order moment of inertia is calculated, the moment vectors must be entered at twice the crank angle (see the right-hand position and moment diagram in Fig. 10.21). Just as with the resultant first-order moment of inertia, the resultant of the maximum values is projected onto the cylinder axis and rotated by 90°.

### 10.1.4 Selected Elements of Reciprocating Machines

#### Crankshaft Drive

The crankshaft drive can be constructed in a plunger piston or crosshead design. The crosshead relieves the piston from lateral piston force, especially in large piston machines (Fig. 10.22).

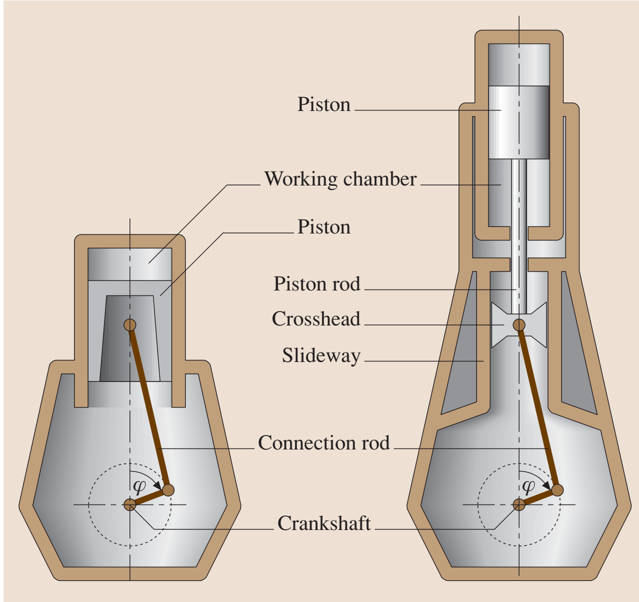



**Fig. 10.22** Plunger piston and crosshead crankshaft drive (after [10.6])

#### Crankshaft

The crankshaft shown in Fig. 10.23 has a shaft journal (1) running in bearings connected by webs (3) to the crankshaft journal. Counterweights (4) on the webs balance the rotating inertial forces. As a rule, the crankshaft is suspended by $z+1$ main bearings, where $z$ is the number of cylinders.

The crankshaft is under stress from forces and bending and torsional moments. The webs are dimensioned accordingly to handle the high stresses.

Different methods are used to manufacture crankshaft blanks. A difference is principally made between cast and forged crankshafts. Cast crankshafts weigh 10% less than forged ones because of the low density of the nodular graphite cast iron frequently used. Giving cast crankshafts a hollow design can boost this even more. A significant disadvantage of cast over forged crankshafts is the low elastic modulus of penetration and associated lower stiffness. In Europe, the market share of cast-iron car and truck crankshafts that are not highly stressed is 60% [10.7].

The most important forging methods can be classified as open die forging and closed die forging. While hammer forging is only used for prototypes and custom-made pieces, closed die forged crankshafts are primarily used in cars and trucks in conjunction with large lot sizes.

If the size of the crankshaft being produced exceeds the production possibilities of forging and casting, the shaft and crankshaft journal are directly connected to one another by the webs. These so-called multi-piece crankshafts are primarily used in the manufacture of large diesel engines.

Forged and cast crankshafts are subjected to different post-processing in order to increase a crankshaft's component strength. Apart from inductive hardening to increase the bearing journal's resistance to wear, rolling the radii in the transition bearing journal and web and nitriding build up residual compressive stresses in the journal and radii zone. As a result, the fatigue strength can be increased considerably in these highly stressed zones.

#### Connecting Rod

The connecting rod transmits forces between the piston and crankshaft journal. It is not only under stress from tension and compressive forces but is also subjected to bending by the respective inertial forces. This necessitates a design that is both rigid and light, especially in high-speed combustion engines.

The connecting rod (Fig. 10.24) consists of a shank and two conrod eyes, which act to connect the piston and the crankshaft. It is cast, die forged or sintered. Heat-treated steel as well as gray cast iron, malleable iron and light metals are used as materials. Fracture separations, also called cracks, separate (straight or

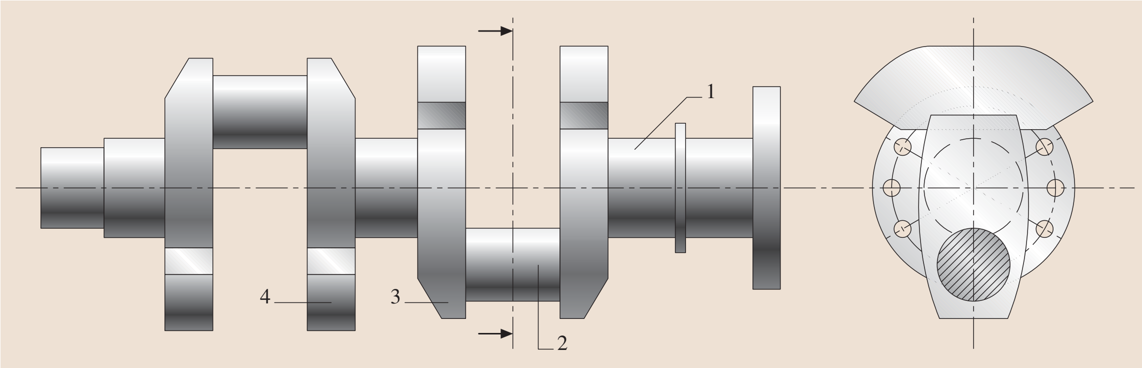



**Fig. 10.23** Elements of a crankshaft (after [10.4])

obliquely) the bottom (large) small-end bearing. Anti-fatigue bolts hold the halves of the large conrod eye together.

### Piston

The piston's job is to transmit fluid forces to the connecting rod or connecting rod forces to the fluid (Fig. 10.25).

In addition, the piston seals the working chamber by means of piston rings. The number of piston rings is dependent on the pressure difference between the upper surface and the bottom surface. In combustion engines, for example, the piston rings dissipate the heat generated in the pistons, which piston cooling can subsequently pass to the motor oil.

Hence, having high resistance to wear and resistance to heat as well being able to conduct heat and being lightweight are the most important requirements on a piston.

This is why pistons are usually made of light alloys and less often gray cast iron (smaller pistons). Low piston weight has a positive effect on the oscillating inertial forces. Steel and cast steel are only used for the top part of multi-piece pistons or for plunger pistons.

### Lubrication

The lubrication of piston machines serves the functions:

- Forming a stable lubricating film between the structural parts
- Reducing friction
- Reducing wear in the machine
- Cooling and cleaning bearings and sliding surfaces
- Forming a seal between the piston ring and cylinder wall

Pressure circulation lubrication is primarily used. The oil is delivered by a pump that is powered by a piston machine or electrically powered. It is purified, cooled and subsequently pumped into the main oil gallery. From there, it reaches the lubricating points such as the crankshaft, valve gear and piston.

### Cooling

The heat losses occurring on the walls of the working chamber necessitate cooling of the corresponding components. The heat flow dissipated is dependent on the surface, the heat transfer coefficients and the temperature difference. Water and air are mainly used as coolants and prevent overheating of the components and lubricant as well as power losses caused by filling losses. However, water's better cooling effect increases the complexity of designing as well as manufacturing of the machine. Water-cooled machines are given a cooling jacket through which water is pumped. In the case of air cooling, the design of

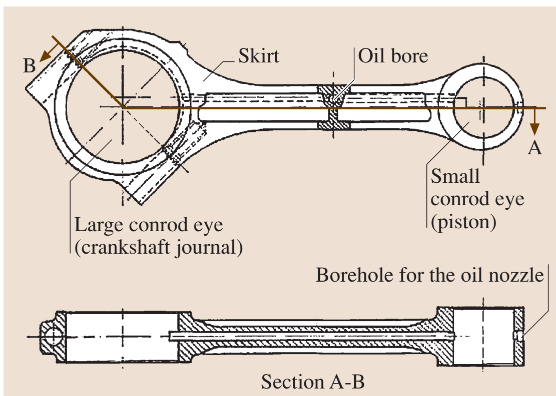



**Fig. 10.24** Connecting rod design (after [10.6])

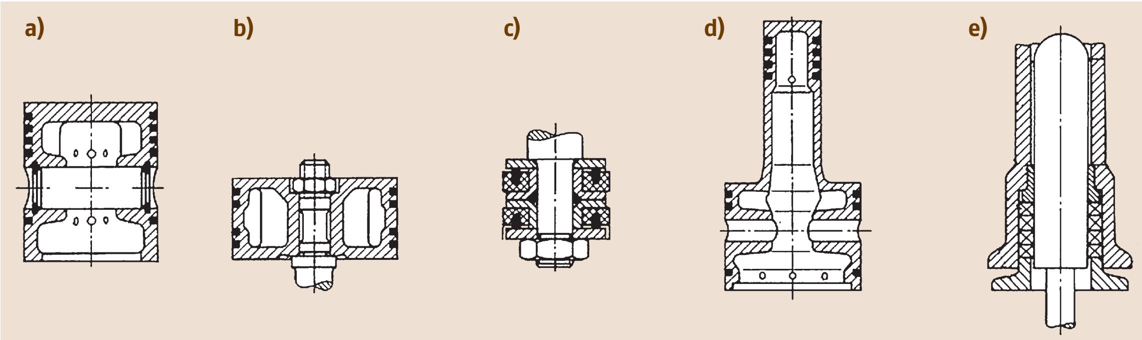



**Fig. 10.25a–e** Piston variants (after [10.4]). **(a)** Plunger, **(b)**, **(c)** disc pistons, **(d)** stepped piston, **(e)** plunger piston

the fins enlarges the surface of the components being cooled.

### Suspension

As a rule, a split bearing is used for the suspension in reciprocating machines because of the offset shape of the crankshaft. Exceptions are multi-piece crankshafts or small and medium-sized piston compressors in which antifriction bearings, among others, can be used.

Their simple design and the small space required in the crankcase, their ability to absorb impacts and vibrations as well as their low mass is increasingly leading to friction bearings with hydrodynamic lubrication being used in reciprocating machines.

Apart from great strength, the materials used must also have optimal tribological properties. Hence, friction bearings are usually designed with a harder matrix (e.g., CuSn, AlCu) into which a soft, low-melting-point material (e.g., Pb, Sn) is incorporated.

The suspension's reliability is crucially important for guaranteeing a piston machine's operation. Hence, optimal engineering utilizes improved bearing materials and state-of-the-art calculation methods.

# 10.2 Positive Displacement Pumps

## 10.2.1 Types and Applications

A positive displacement pump is a work machine for incompressible media (fluids), the working chamber of which is periodically altered by the displacer (piston). The displacer's motion can be either oscillating (reciprocating pump) or rotating (rotary piston pump).

Reciprocating pumps mainly use automatically operating, pressure-controlled valves to control the process, i. e., to connect and disconnect the working chamber to the suction and the pressure line. Rotary piston pumps use ports to do this.

In each case, the discharge of the medium directly follows the characteristic of the displacer's motion (volume displacement).

Theoretically, the maximum pressure achievable with positive displacement pumps is unlimited and determined only by internal leakage losses, the medium's compressibility, the component strength and the motive power.

During delivery, mechanical energy is transferred to the medium as potential energy, which serves to compensate for level or pressure differences.

A difference is made between reciprocating pumps with rigid pistons (disc and plunger pistons and plungers, Fig. 10.26) and with elastic pistons (diaphragm, hose, Fig. 10.27).

Radial and axial piston pumps in which the piston is typically designed as a plunger are also used in high-pressure hydraulics. Depending on the design (radial or axial), the stroke motion in these pumps is generated by a rotating eccentric cam or a swash plate and a fixed cylinder unit, i. e., similar to reciprocating pumps with crankshaft drive.

In part, the lift stroke for a fixed eccentric or swash plate is also produced with a rotating cylinder unit (drum/star). In systems with a rotating cylinder unit, control blocks with ports are used instead of the working valves otherwise customary in reciprocating pumps (Figs. 10.28, 10.29).

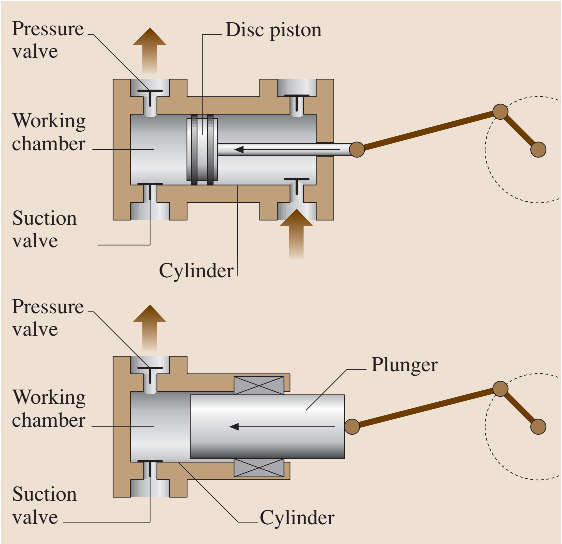



**Fig. 10.26** Rigid piston design

Rotary piston pumps are classified according to the primary motion of the displacer as single rotation piston machines, planetary rotation piston machines and circulation piston machines (Figs. 10.30–10.33).

Other designs for delivering media with solid fractions or for liquefied gases or even with direct drive without crankshaft drive, e.g., steam and pneumatic pumps or hand pumps, are characterized in every configuration by special requirements and have different, usually oscillating types of motion, which for the most

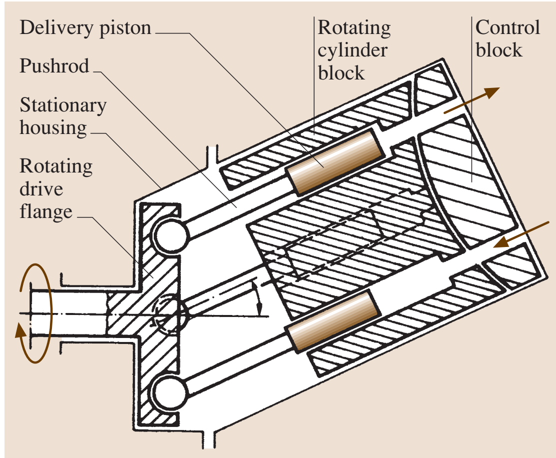



**Fig. 10.28** Axial piston pump

part are directly generated without a crankshaft drive or the like (Fig. 10.34).

Furthermore, other designs exist that cannot be directly classified in the categories cited because of their types of motion. One example is the peristaltic pump used in medicine and laboratories (Fig. 10.35).

The vane pump, primarily used for garden irrigation and in small boats, is simple in design and durable in operation (Fig. 10.36).

## 10.2.2 Basic Design Parameters

Figure 10.37 presents the principle of reciprocating pump operation.

The crankshaft drive, consisting of the crankshaft, connecting rod and crosshead, converts rotary drive mo-

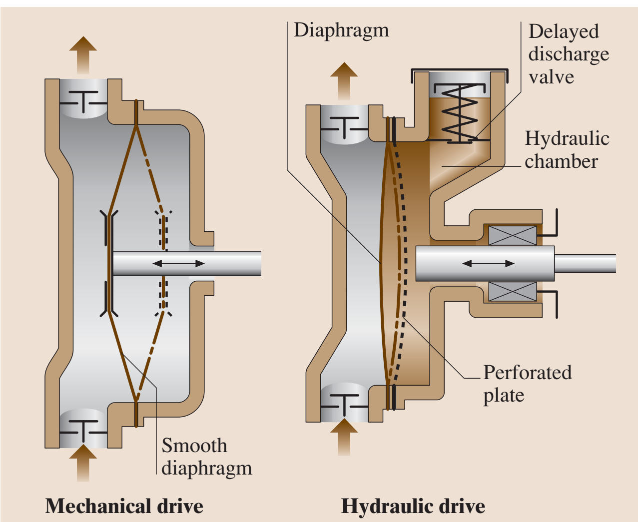



**Fig. 10.27** Diaphragm designs

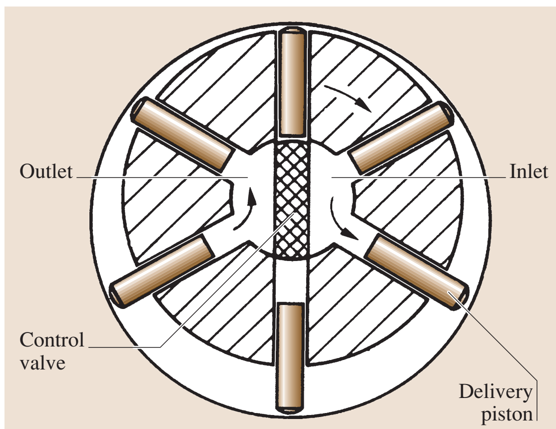



**Fig. 10.29** Radial piston pump

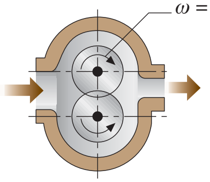
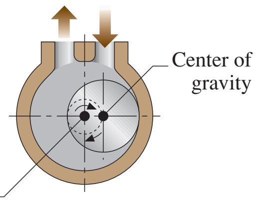
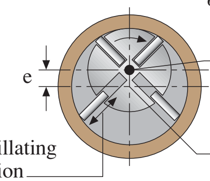
| Single rotation piston machines | Planetary rotational piston machine | Circulation piston machine |
|---|---|---|
| Moving parts induce uniform rotation around their centers of mass. | The center of gravity of the uniformly rotating displace describes a circular path. | Along with uniformly rotating parts there are also irregularly moving parts that execute circular or circle-like or even oscillating motions. |
| Moving parts are directly balanced: bearings and shafts are not subjected to stress from centrifugal forces → High speeds are possible. | The rotary imbalance can only be directly balanced. Increased stresses on bearings caused by centrifugal forces. Speeds are limited. | Increased friction in the machine → overall efficiency decreases. Wear increases. |
| Examples: Gear pump, screw design, roots blower, gear tooth design, liquid ring design. | Examples: Trochoidal-type design, eccentric screw pump, eccentric design, rolling piston design. | Examples: Multi-cell or impeller cell design, rotary valve design, external-vane design. |
| Operating principle   (Gear pump) | Operating principle   (Eccenter-type design) | Operating principle   (Vane-type design) |

**Fig. 10.30** Rotary piston machines

tion into an oscillating motion, which the piston rod transmits to the piston. In the cylinder, the piston executes a motion between the two dead center positions, the crank dead center (CDC) and the head dead center (HDC), whereby the volume trapped between piston and cylinder cover diminishes and expands periodically.

In principle, the function and process are identical for rotary positive displacement pumps. The resultant working cycle can be broken down as follows:

1. Induction. The volume of the working chamber expands during the piston's motion from CDC to HDC. This produces a vacuum in the working chamber causing the suction valve (SV) to open. This vacuum causes the fluid delivered from the suction line to flow into the cylinder. At nearly constant pressure, induction theoretically occurs during the complete piston stroke (corresponding to line 1–2 in the $p$–$V$ diagram).
2. Pressure rise. The piston motion reverses in the crank dead center. The resultant reduction in volume causes the pressure in the working chamber to rise and the suction valve to consequently close. Since liquids are theoretically incompressible, the pressure rises at a constant volume to the pressure at the pressure valve (PV) (corresponding to line 2–3 in the $p$–$V$ diagram).
3. Expulsion. The reduction in volume during the piston's motion from CDC to HDC causes the pressure in the working chamber to increase. When a pressure is attained, which dominates on the pressure side of the pump, the pressure valve opens. The fluid is expelled approaching the pressure, which dominates at the outlet (corresponding to line 3–4 in the $p$–$V$ diagram).
4. Pressure drop. When the head dead center is reached, the piston motion reverses again so that the pressure drops at constant volume causing the pressure valve to close and the suction valve to open (corresponding to line 4–1 in the $p$–$V$ diagram).

Whereas the crankshaft has a constant angular velocity (speed), the piston speed in the cylinder is not constant. It is zero at the dead centers. During its motion from one dead center to the other, there is acceleration and deceleration, each corresponding to the crankshaft's law of motion (Sect. 10.1.3). The piston speed is a function of the angle of rotation $\alpha$ and, as a function of the crankshaft drive's connecting rod ratio (10.23), has an approximately sinusoidal characteristic line.

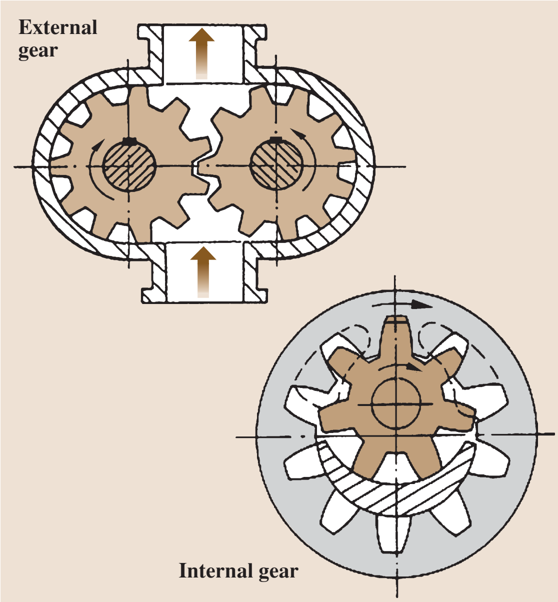



**Fig. 10.31** Single rotation piston design

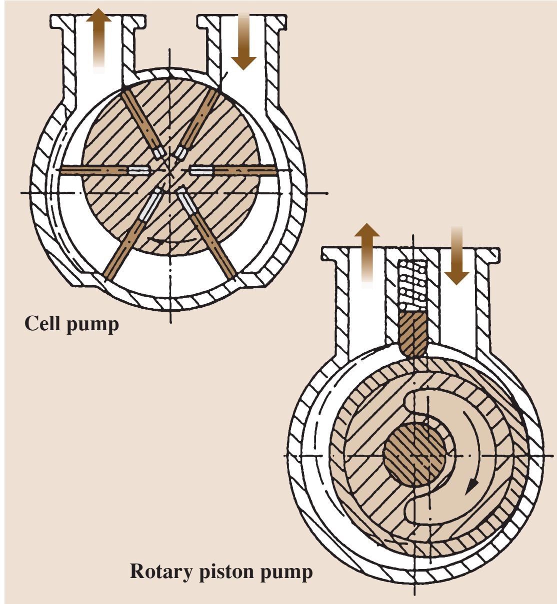



**Fig. 10.33** Circulation piston design

### Operating Behavior

The $p$–$V$ diagram in Fig. 10.37 represents an ideal development of pressure, presuming a massless fluid flows without loss. Since losses and inertial forces always have an effect under real conditions, the real $p$–$V$ diagram of a piston pump in Fig. 10.38 is produced. The ideal pressure development from Fig. 10.37 has been incorporated for comparison.

- Since the frictional losses in the suction line and the suction valve have to be compensated for, suction occurs at a pressure lower than the suction tank pressure. Moreover, another drop in pressure is needed to open the suction valve since the valve opening resistance (valve acceleration, surface ratios at the valve seat, valve elastic force) has to be overcome. The mass inertia of the decelerated liquid column or the suction pressure can also cause a slight rise in pressure toward the end of the intake stroke.
- Since the mass inertia does not cause the suction valve to close abruptly when the CDC is reached and a real fluid is not fully compressible, the pressure does not rise at a constant volume. Moreover, fluids frequently contain small quantities of gas that cannot be compressed. As a result, the real pressure rise from 2′ to 3′ is not isochoric.
- Expulsion occurs at a higher pressure than on the pressure side of the pump since the corresponding losses again have an effect here. Analogous to the

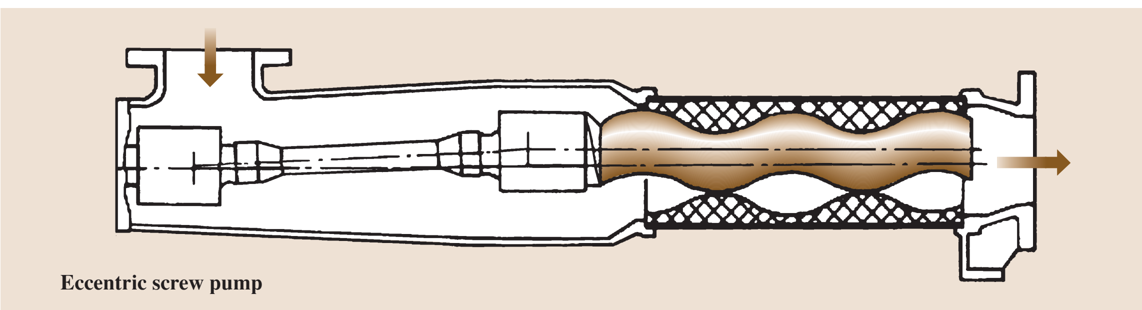



**Fig. 10.32** Planetary rotation piston design

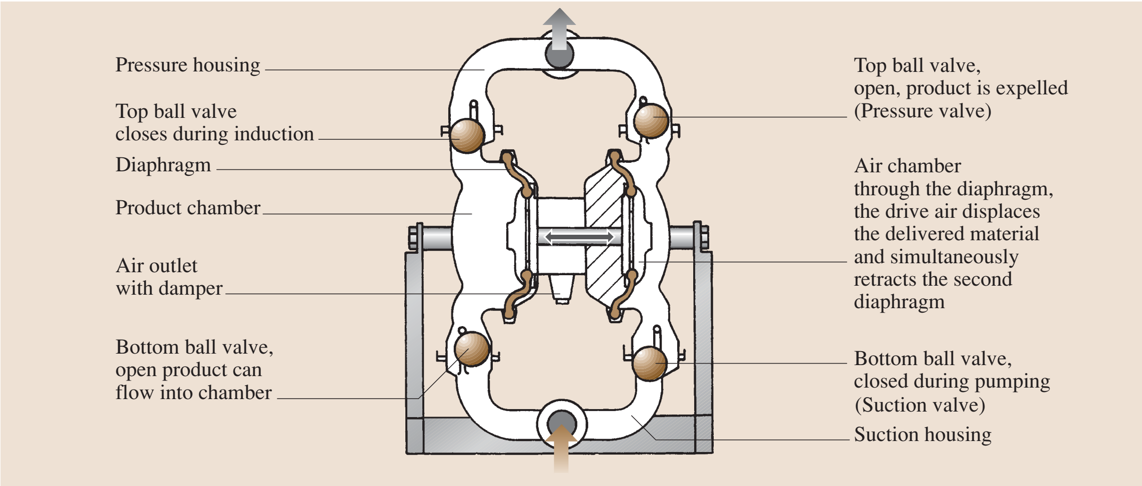



**Fig. 10.34** Compressed-air diaphragm pump (after [10.8])

suction valve, an additional increase in pressure is needed to open the pressure valve.

- In turn, the pressure cannot drop at a constant volume since there are inertial forces and potential back-flows from the pressure line when the pressure valve closes and the gas fractions re-expand. As a result, the ideally isochoric pressure drop from $4'$ to $1'$ is not isochoric.

In contrast to centrifugal pumps, piston pumps have a characteristic curve with a delivery rate that is nearly independent of delivery pressure. As a result of the displacement, the delivery rate is calculated as the product of the piston surface $s$, stroke and rotational speed $n$

$$\dot{v} = Q = A_{\mathrm{K}} s n \,. \tag{10.48}$$

Consequently, throttling, i. e., changing the system head curve, cannot regulate the delivery rate. On the other hand, an inordinate rise in pressure has to be prevented from destroying the pump and system components.

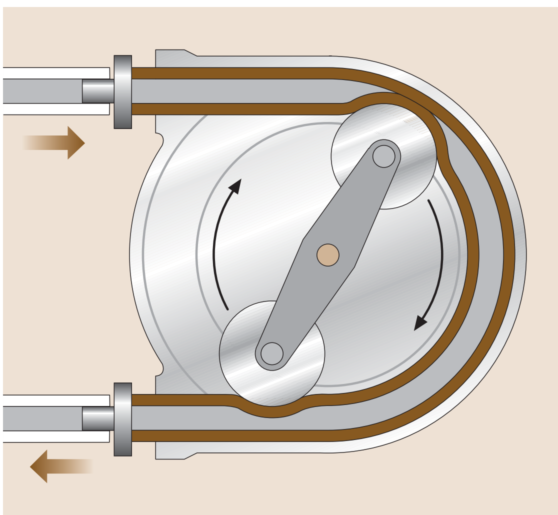


**Fig. 10.35** Peristaltic pump (after [10.9])

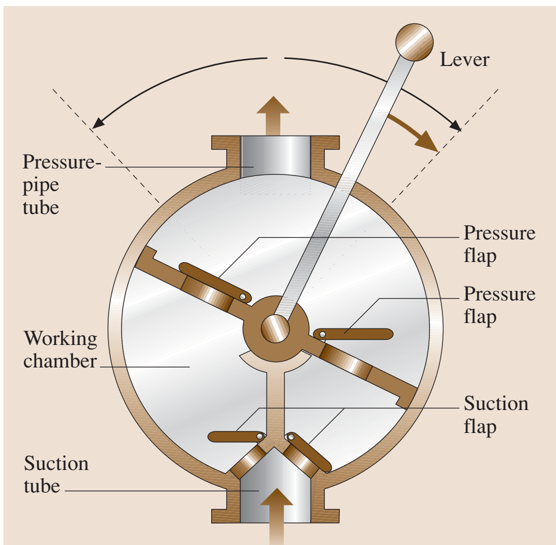



**Fig. 10.36** Sliding vane pump

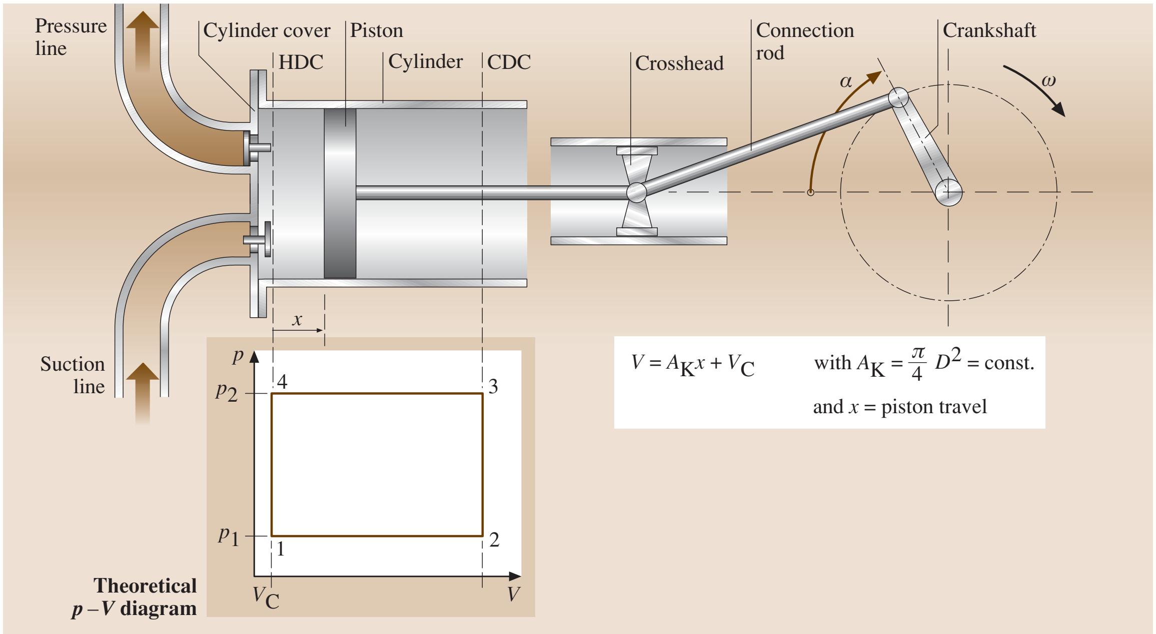



**Fig. 10.37** Principle of reciprocating pump operation

Hence, any time a piston pump is used, a *safety valve* must be incorporated on the system's pressure side.

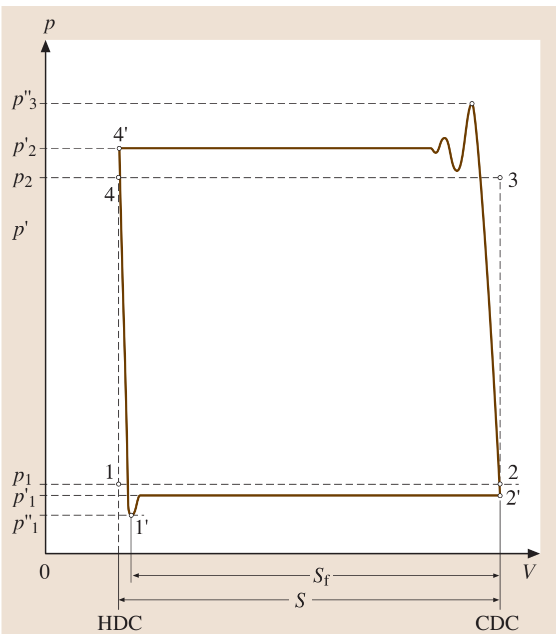



**Fig. 10.38** Ideal and real piston pump $p$–$V$ diagram

### Energy and Efficiency

***Specific Effective Energy of the Complete System $w_{st}$.*** Figure 10.39 shows a simple pump system.

The pump feeds mechanical work to the fluid in order to increase the pressure of the potential energy, i. e., to facilitate delivery and/or a rise in pressure.

If this work is applied to the fluid's mass (disregarding changes in density), then the specific effective energy of the system is obtained

$$w_{\mathrm{st}} = w_{\mathrm{geo}} + \frac{(p_{\mathrm{a}} - p_{\mathrm{e}})}{\rho} , \tag{10.49}$$

$$w_{\mathrm{geo}} = H_{\mathrm{geo}} g , \tag{10.50}$$

where $w_{\mathrm{geo}}$ is the specific geodetic delivery work, $p_{\mathrm{e}}$ is the pressure on the fluid in the suction tank, $p_{\mathrm{a}}$ is the pressure on the fluid in the pressure tank, $\rho$ is the density of the pumped medium, $g$ is the gravitational acceleration, and $H_{\mathrm{geo}}$ is the level difference.

Expressed in pressures and levels, the following ensues

$$p_{\mathrm{st}} = H_{\mathrm{geo}} \rho g + p_{\mathrm{a}} - p_{\mathrm{e}} , \tag{10.51}$$

$$H_{\mathrm{st}} = H_{\mathrm{geo}} + \frac{(p_{\mathrm{a}} - p_{\mathrm{e}})}{\rho g} . \tag{10.52}$$

The system's specific effective energy is independent of the type of pump, the size of the delivery flow, and the layout of the pipe system.

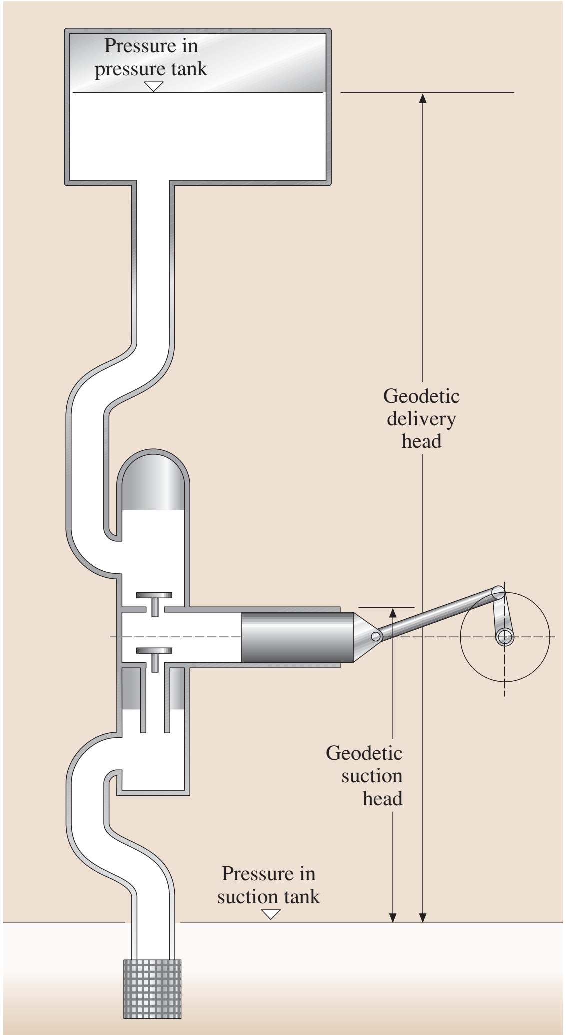



**Fig. 10.39** Piston pump system

*Specific Effective Energy of the Pump w.* The specific delivery work required to deliver the mass flow of the fluid in the selected system is referred to as the specific effective energy of the pump. The proportions of energy loss and velocity energy are taken into account here

$$w = w_{st} + w_{dyn} + w_V \,, \tag{10.53}$$

$$w_{dyn} = 0{,}5\left(v_2^2 - v_1^2\right) \,, \tag{10.54}$$

where $v_2$ and $v_1$ are the velocities in the suction and pressure tubes, respectively, and $w_v$ is the specific energy loss in the suction and pressure lines.

The pump's specific effective energy depends on:

1. The type of pump
2. The size of the delivery flow and
3. The layout of the pipe system

**Delivery Rate $\eta_{Vol}$.** As a result of volume losses, a pump's effective delivery rate is lower than the theoretical rate. These losses are caused by:

1. Leaks or valve-closing delays (volumetric efficiency $\lambda_L$)
2. The presence of gases or incomplete filling ($\lambda_F$)

The product of $\lambda_L$ and $\lambda_F$ constitutes the delivery rate $\eta_{Vol}$. The following empirical values apply to the delivery rate $\eta_{Vol}$:

1. 0.88–0.92 for small pumps
2. 0.92–0.96 for medium-sized pumps
3. 0.96–0.98 for large pumps

**Indicated Efficiency $\eta_i$.** Indicated efficiency expresses the ratio of a system's effective power to a pump's indicated power (determined from the indicator diagram). Thus, hydraulic losses are also taken into account.

$$\eta_i = \frac{P_Q}{P_i} \,, \tag{10.55}$$

$$P_Q = \rho Q w_{st} \,, \tag{10.56}$$

where Q is the feed rate and $P_i$ is the indicated power (from the indicator diagram).

**Mechanical Efficiency $\eta_m$.** Mechanical efficiency expresses all the losses caused by *mechanical friction* inside the pump. In the process, the ratio of indicated power to coupling power (input power) is expressed by

$$\eta_m = \frac{P_i}{P_{zu}} \,. \tag{10.57}$$

Experience has shown that pumps with crankshaft drive have a mechanical efficiency in the range

$$\eta_m = 0.85 - 0.95 \,. \tag{10.58}$$

**Total Efficiency of the Pump $\eta$.** Total efficiency $\eta$ is expressed by the ratio of the effective power to the input power

$$\eta = \frac{P_Q}{P_{zu}} \,. \tag{10.59}$$

Experience has shown that a pump with crankshaft drive has a total efficiency in the range of $\eta =$ 0.70–0.80.

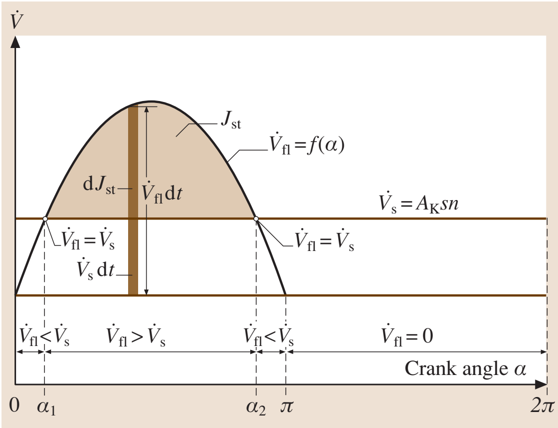



**Fig. 10.40** Time progression of the volumetric delivery of a single-action one cylinder pump

### Mass Actions

On the one hand, the reciprocating pump's discontinuous working process results in a constantly fluctuating speed of the pumped medium in the connected lines. On the other hand, the portions of the pumped medium subjected to accelerations undergo a mass action. The time-varying delivery rate follows the laws of piston motion up to a fixed limit (the vapor pressure) so that the time progression of the volumetric delivery corresponds exactly to the piston velocity. The valves' action is used to allocate the positive and negative fractions to the intake and the delivery stroke (Fig. 10.40).

The crankshaft's rotation is split into the intake and the delivery stroke. Thus the total delivery rate must be pumped during a half rotation of the crankshaft. The maximum value of the speed is considerably larger than the average value. There are only two times during both the intake and the delivery stroke at which the instantaneous delivery rate corresponds exactly to the average volumetric flow. This is not the case at any other time.

The maximum acceleration is at a crank angle of 0°, i. e., the theoretical start of the intake stroke. However, a maximum acceleration of the amount of fluid also corresponds to a maximum mass action, which results in a maximum reduction in the pressure in the working chamber (Fig. 10.41).

To prevent cavitation, the so-called vapor pressure $p_0$ of the fluid must not be reached during pump operation. Consequently, the piston acceleration and the mass of the fluid present in the intake tract yield the

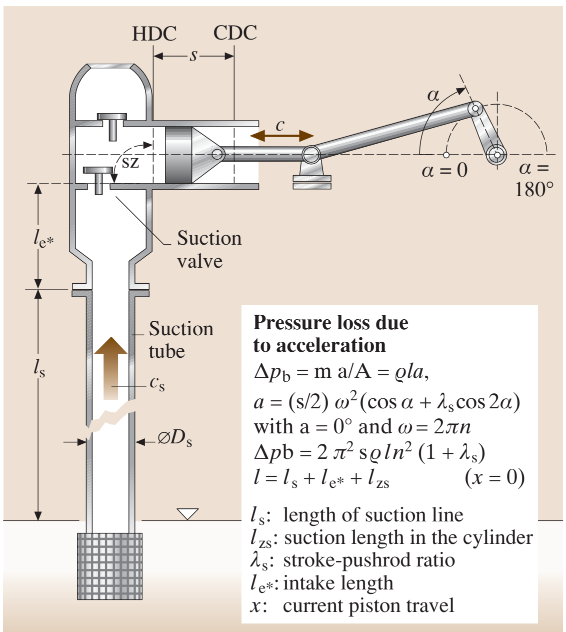



**Fig. 10.41** Suction side of a reciprocating pump system

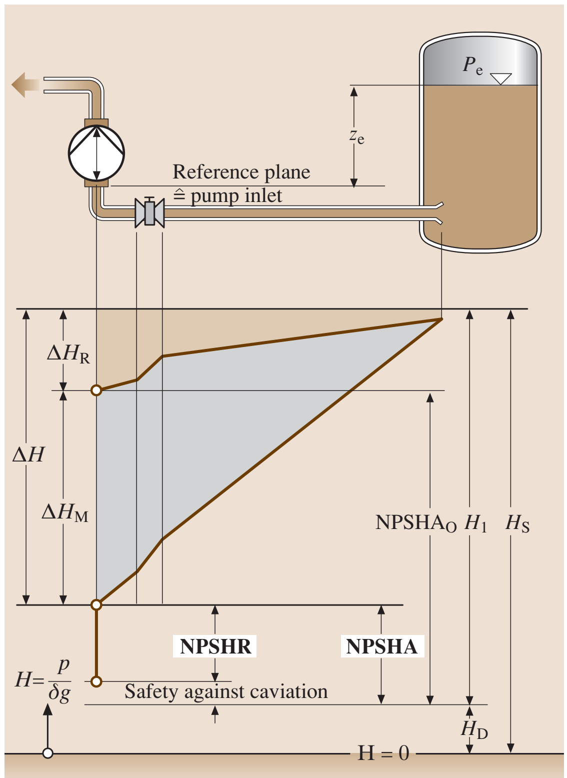



**Fig. 10.42** NPSH value of the intake tract of a pump system (after [10.10])

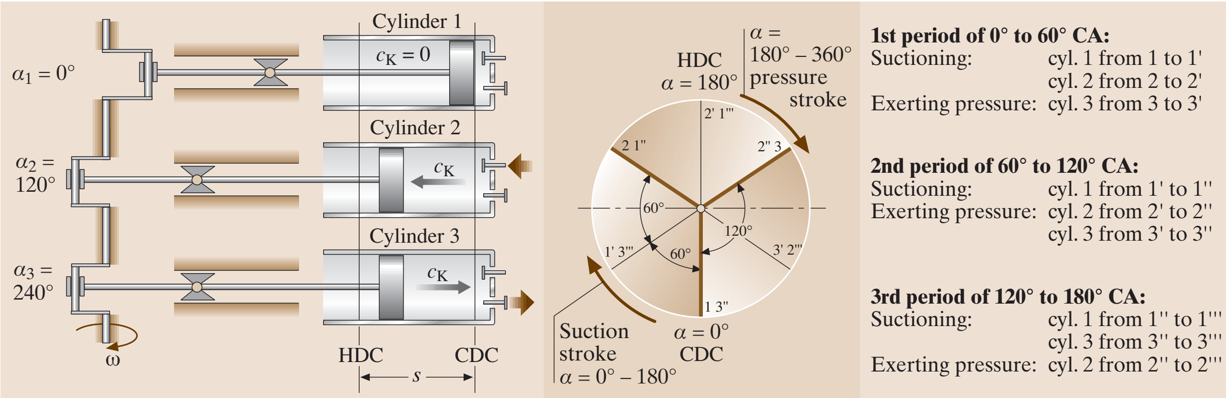



**Fig. 10.43** Function of a three-cylinder pump

maximum permissible suction head. Conversely, when the suction head is known, it yields the maximum permissible speed for the pump.

The net positive suction head (NPSH; Fig. 10.42) is used to designate a pump's volume flow rate. The difference between the actual and the required NPSH is a measure of protection against cavitation.

When several working cylinders are employed, conditions change with the effect that several cylinders are able to draw from one suction line at the same time (Fig. 10.43). Here, the drop in pressure as a result of the mass action must be determined individually for the single parts of the intake tract. Employing several cylinders improves the overall balance since the entire amount of fluid is not subjected to maximum acceleration and, when there are upwards of three working chambers, the flow velocity in the suction line is never zero.

To reduce the pulsation of the delivery rate both on the suction and the pressure side, employing several cylinders is expedient.

If using several cylinders proves insufficient, then installing elastic components, the so-called expansion chamber, is advisable. Here, interconnecting an elastic accumulator (gas cushion) achieves a homogenization of the delivery rate.

The cyclic irregularity of the the pressure on the piston is consulted as a decision criterion for integrating an expansion chamber

$$\delta_p = \frac{(p_{K-max} - p_{K-min})}{p_{K-m}}\,, \tag{10.60}$$

where

$$p_{K-m} = \frac{p_{K-max} + p_{K-min}}{2}\,. \tag{10.61}$$

This applies to both the suction and the pressure side.

The cyclic irregularities considered limit values for the use of expansion chambers are $\delta_{p-s} \leq 0.1–0.05$ (suction side), and $\delta_{p-d} \leq 0.05–0.02$ (pressure side).

The expansion chambers can be constructed both as a simple air tank open toward the fluid (Fig. 10.44) or as a diaphragm or bladder accumulator (Fig. 10.45).

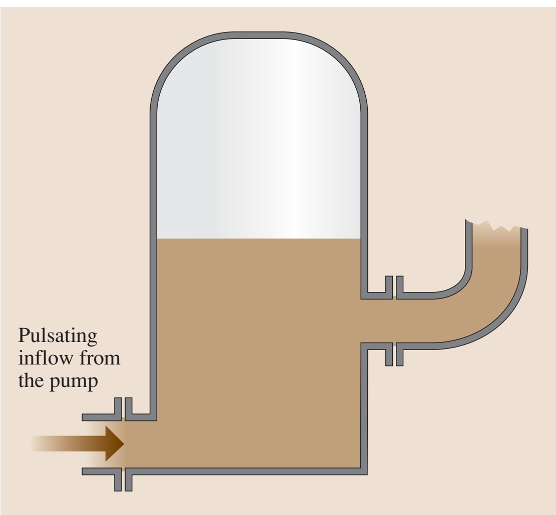



**Fig. 10.44** Pressurized air chamber

## 10.2.3 Components and Construction of Positive Displacement Pumps

### Working Valves

Automatically operating, pressure controlled valves are used in reciprocating pumps to separate the suction and expulsion processes. These valves can be constructed as

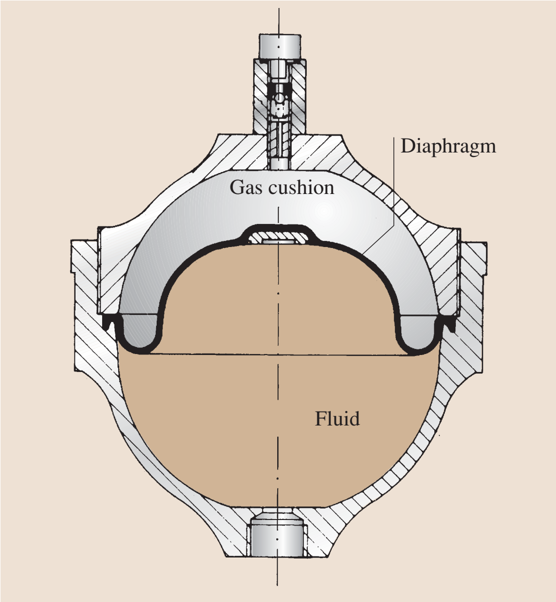



**Fig. 10.45** Diaphragm accumulator (after [10.11])

flat seat, conical seat or ball valves. Both, elastic force and gravitational force can trigger their closing forces (Fig. 10.46).

Multiple annular valves (Fig. 10.47) are also used to reduce the force of acceleration at the valve and above all to lower their speed when they close. These make large openings possible when strokes are small.

To open such a valve, a pressure differential must always be present, which essentially depends on the elastic force, the friction and the valve face (Fig. 10.48).

### Reciprocating Pumps

*Compression Pumps.* Compression pumps are pumps that, as boiler feed pumps for example, work approaching pressures up to 7000 bar and achieve delivery flows of 0.01–160 $m^3/h$. As a result, pump components are under most extreme stresses, most notably in the cylinder (called the pump head here) the valves and the plunger seal (stuffing box). In view of the high pulsating stress of the components, the design not only has to apply appropriate methods of calculation and employ the requisite material but also produce particular surface finishes to ensure sufficient operational reliability.

For economic reasons, such pumps, which are usually only manufactured in small quantities, are often engineered based on *standardized* engines with crossheads. Thus, they are then designed for specific forces along the rod (piston forces). When standard engines are used for different pumps, a specific maximum piston diameter is produced for every pressure to be reached. The number of cylinders and the speed determine the delivery flow pumped. By combining pump heads in these engines, appropriate pumps can be assembled (modular system) for the widest variety of applications with relatively little effort.

The design engineering and material selection for individual components such as the plunger, pump cylinder, valves, etc. are extremely important for pumps with higher pressures. Composite materials or prestressed components are used at extreme pressures (Fig. 10.49).

An important aspect of design is making the disassembly of wearing parts (valves, stuffing box packing) as easy as possible. Above all, it is essential to avoid having to disassemble the suction and pressure lines when removing the valve since these are very rigid at extreme pressures.

The stuffing box packings in compression pumps are under extreme stress and therefore may not be used as a plunger guide. To this end, a corresponding guide bushing (not functioning as a seal) is used.

*Metering Pumps.* The chemical industry as well as the food processing industry needs metering pumps on a large scale to ensure fluid is metered precisely and consistently. Delivery pressures of up to 4000 bar and precisely repeatable metered quantities of 0.001 l/h or 0.2 mg/stroke up to 200 000 l/h have to be achieved. The required metered quantities are regulated by adjusting the speed or more frequently the plunger stroke. While continuously variable transmissions or frequency converters are widely used to adjust the speed, different variants are used to adjust the piston stroke, which, depending on their design, operate with the *fixed mean position of the piston* or with the *fixed dead center of the piston* from stroke zero onward.

Both plunger- and diaphragm-type metering pumps are constructed.

The stroke can be adjusted in different ways, e.g.:

- Crankshaft drive with adjustable crank radius (Fig. 10.50).
- Double eccentric with adjustable eccentricity (Fig. 10.51).
- Radially adjustable rotary eccentric (Fig. 10.52).
- Spring cam drive units (Fig. 10.53) are often used for simple, primarily smaller diaphragm metering pumps. An adjustable limiting bolt mechanically adjusts the stroke. The stroke can be adjusted by using the limiting bolt to increase or decrease the

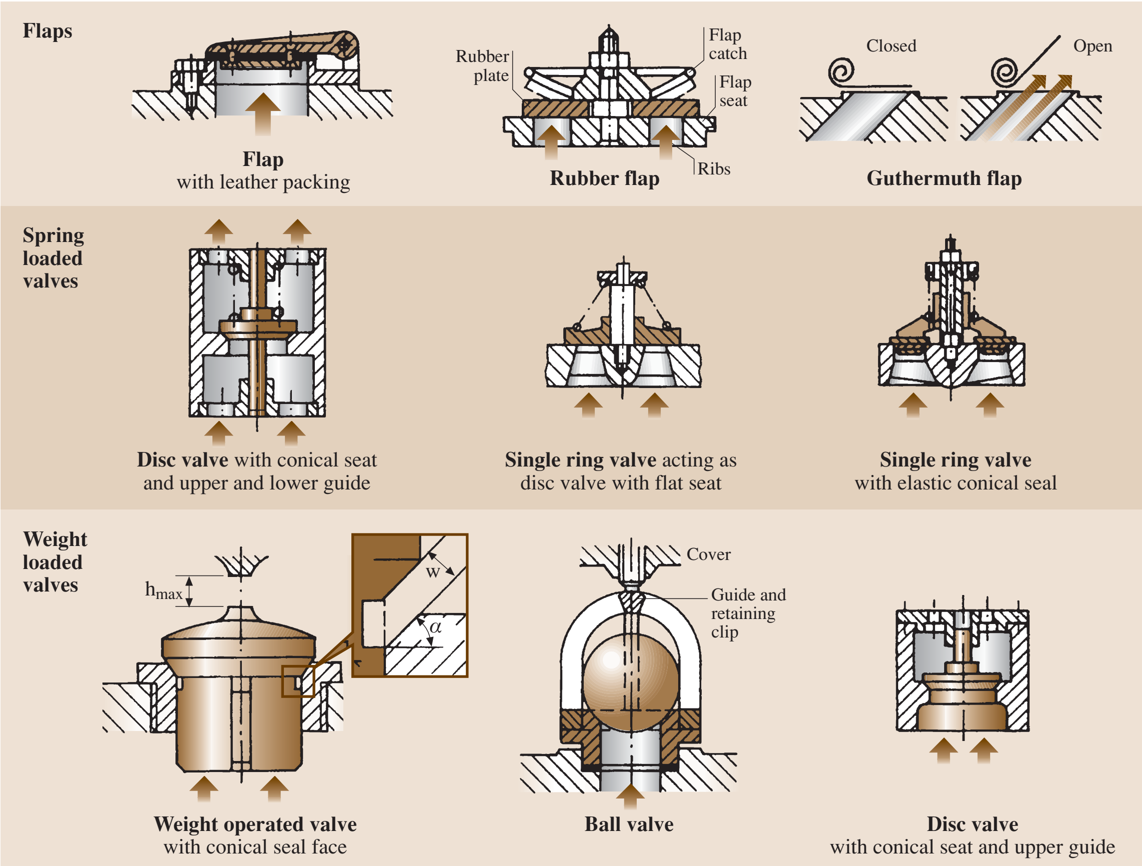



**Fig. 10.46** Valve types (after [10.12])

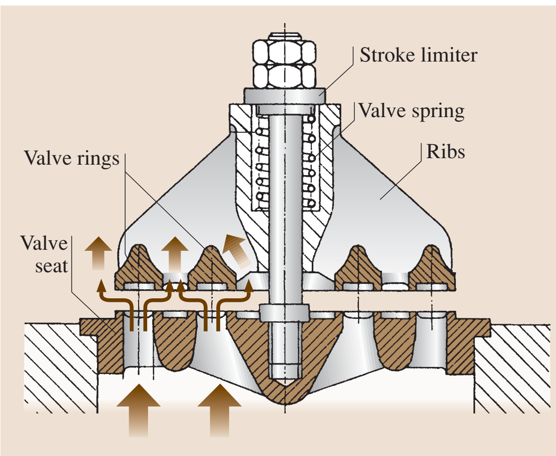



**Fig. 10.47** Multiple annular valves (after [10.12])

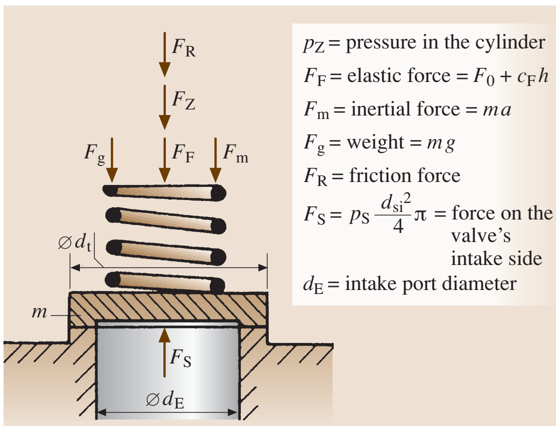



**Fig. 10.48** Forces at the suction valve

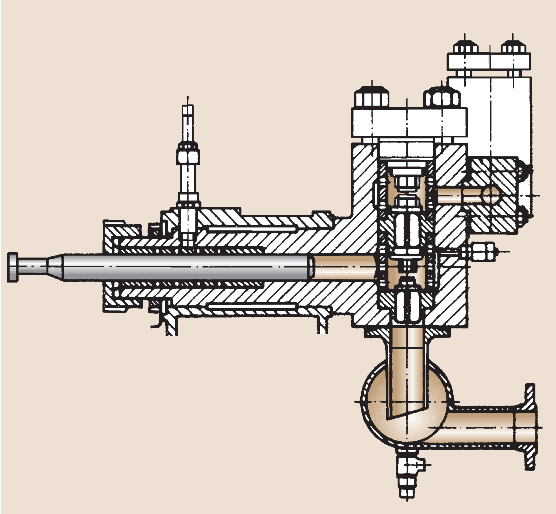


**Fig. 10.49** Maximum pressure piston pump for pressures up to 7000 bar (after [10.13])

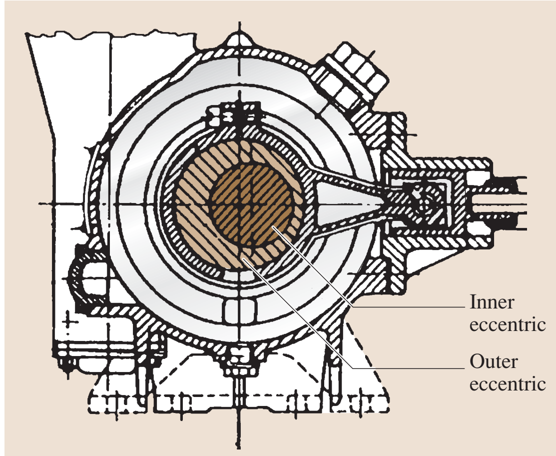



**Fig. 10.51** Double eccentric (after [10.14])

distance between the follower and the cam's basic circular path. A rotary eccentric cam is the cam shape used. A so-called phase angle occurs in which only part of the stroke is utilized. However, strong forces of acceleration act in this drive unit at the start of the effective stroke so that the range of adjustment at higher pressures is limited to values between a complete stroke and a maximum of 50% of the stroke.

Diaphragm metering pumps are used for applications in which the delivery chamber must be absolutely impermeable to the environment. The diaphragm can be operated mechanically as well as hydraulically (Figs. 10.54 and 10.55).

The delivery pressure in mechanical drives subjects the membrane edge to great stress. A hydraulic drive does not subject the diaphragm to great stress and considerably higher delivery pressures are possible.

With regard to adjusting the stroke, the same applies to hydraulically powered diaphragms as to mechanically powered diaphragms. The hydraulic system is

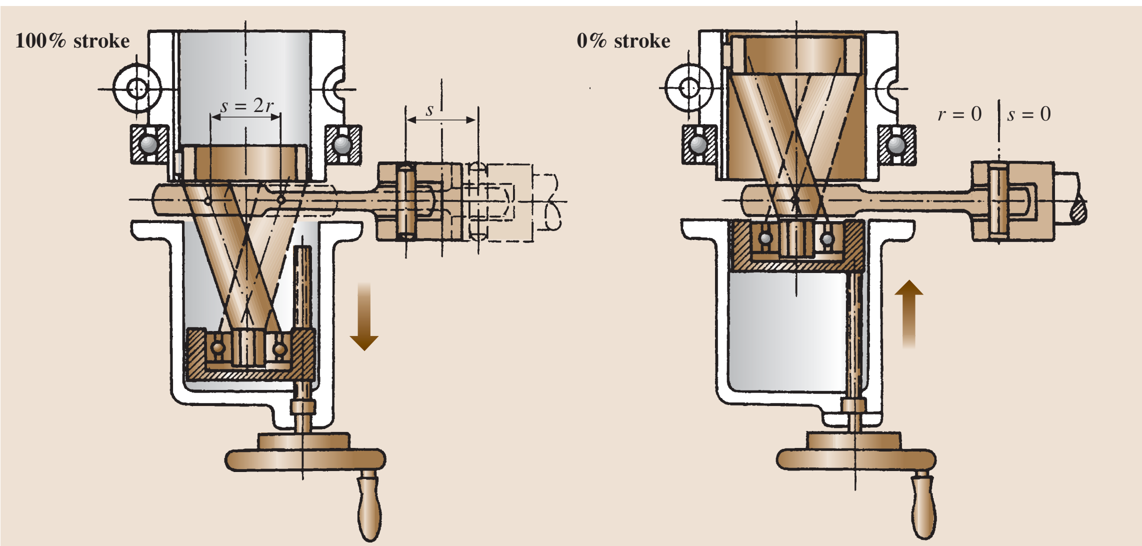



**Fig. 10.50** Adjustable crank radius (after [10.11])

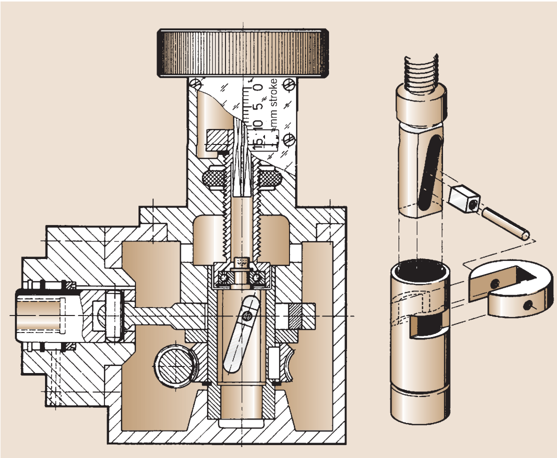



**Fig. 10.52** Adjustable rotary eccentric (after [10.14])

simply connected between the plunger and the diaphragm and serves to transfer the quantity pumped in a ratio to the volume of 1 : 1. The force and travel ratios follow the continuity equation. The amount of leakage potentially occurring in the hydraulic system is replenished from the storage tank during each stroke so that the diaphragm's stroke motion is retained with its end positions. Frequently, there is a diaphragm position control. Employing a device to redefine the diaphragm's end position during each stroke prevents overloading of the diaphragm.

A multiple membrane with an inserted control medium is used for cases in which contamination of the pumped fluid by the hydraulic fluid must be prevented or emergency operation guaranteed.

### Rotary Piston Pumps

Along with the type of motion, the absence of working valves is a significant feature of piston pumps with rotary displacers. Ports control the suction and pressure

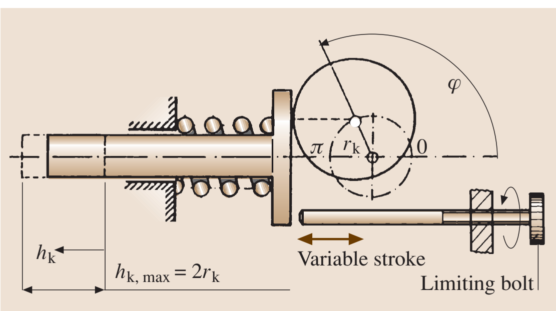



**Fig. 10.53** Flexible cam drive unit

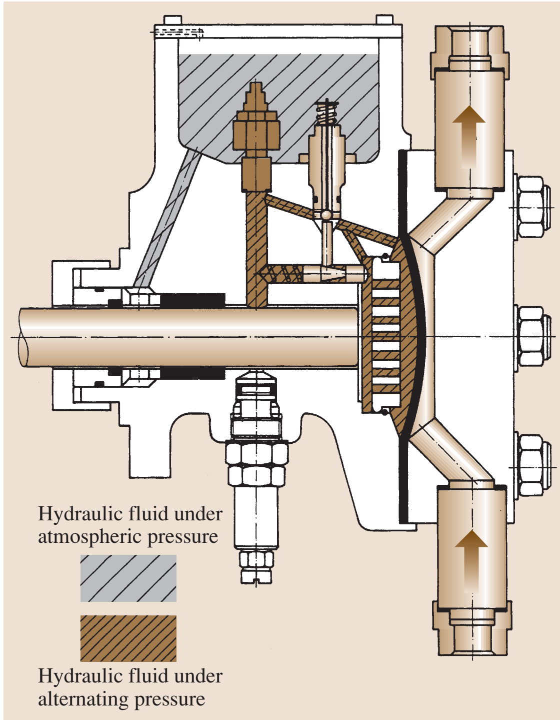



**Fig. 10.54** Diaphragm metering pump with diaphragm position control (after [10.11])

process. As a result, when a pressure difference exists in the adjacent medium, these machines can also function as power machines. In order to prevent this, non-return valves (that do not have the function of working valves) are used.

#### *Single Rotation Piston Machines.*

**Gear Pumps.** A gear pump consists of two or more gearwheels arranged in a housing and intermeshed. The driveshaft is attached outside the housing, the remaining gearwheels being driven by the gearing. The suction and pressure sides are separated by the meshing and the sealing gap between the tooth tips, gear-side surfaces and housing.

The rotating gearwheels cause the pumped medium to be delivered from the suction side to the pressure side. In the zone of gear meshing, the teeth displace the medium from the tooth gaps (displacement effect). Simultaneously, the intermeshed tooth flanks seal the suction chamber off from the pressure chamber (Fig. 10.56 (Sp)). Moreover, the tooth tips seal the individual volumes off from the housing (Fig. 10.56). In addition, the axial gap must be confined to narrow lim-

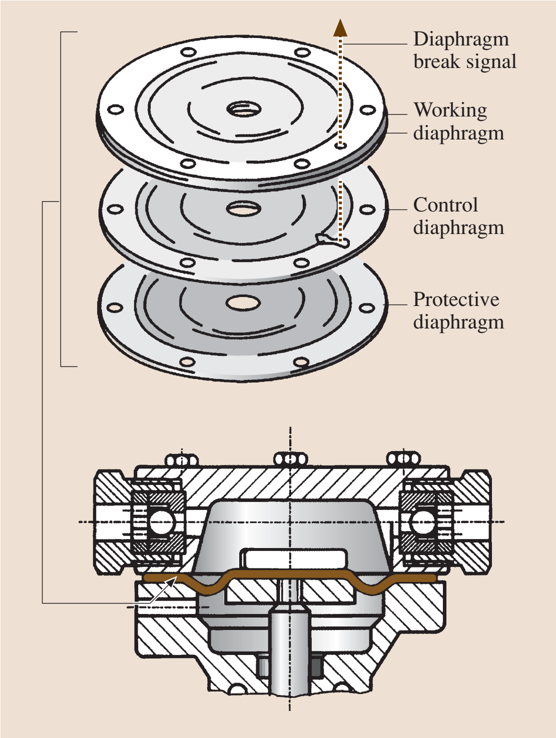



**Fig. 10.55** Multi-diaphragm pump (after [10.11])

its by maintaining slight axial clearances. An optimal seal enables the gear pump to attain high delivery rates (volumetric efficiencies).

As a rule, the same involute profile used for transmission gears is used for the tooth flank profile of gear

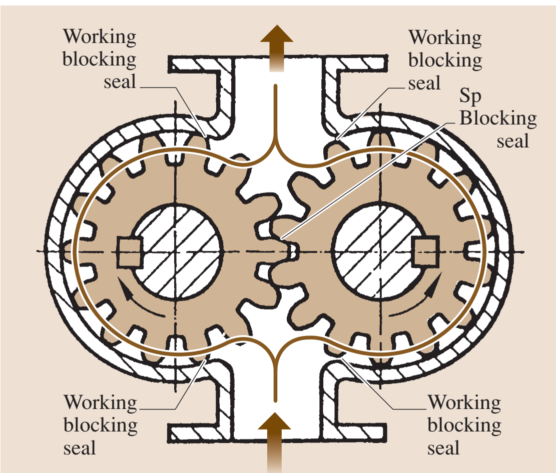



**Fig. 10.56** Principle of the gear pump

pumps. Thus, the gearwheels can be manufactured cost effectively by using the same tools on identical machines. Other, more-expensive profiles are reverted to only for special applications such as pulsation-free sine pumps. Helical gears are also rarely used. While intended to achieve smoother running, they generate an axial thrust to the gearwheels and bearings as well as unequal axial gap widths.

When rolling, a self-contained volume is produced in the tooth gap into which the opposing tooth immerses. This causes a change of volume. Since a change of volume in enclosed spaces cannot be tolerated when fluid is being pumped, appropriate measures must ensure fluid is supplied to and removed from the tooth gap (Fig. 10.57).

To this end, either so-called compression slots are made in the side panels of the pump or on the teeth flanks or boreholes are made in the bottom land of a gear, which the grooves in a fixed axis connect to the suction or pressure chamber. Alternatively, the tooth flank clearances are made sufficiently large.

When they are being engineered, the clearances between the components must be dimensioned to correspond to the viscosity of the pumped medium and the delivery pressure. This can lead to problems, most notably when there is axial play.

Axial hydraulic clearance compensation is applied to improve impermeability at high pressures and to simultaneously reduce friction during starting. Thus, delivery pressure is utilized to reduce running clearance.

Internal gear pumps can additionally be given radial hydraulic clearance compensation (Fig. 10.58).

Low-pressure pumps ($\Delta p < 16\,\text{bar}$) are usually designed with eight to 14 teeth while medium- to high-pressure pumps ($\Delta p > 160\,\text{bar}$) have 16–25 teeth.

**Single Rotation Piston Pumps.** Rotational motion piston pumps are often incorrectly called eccentric rotational motion piston pumps. Since they are frequently positively driven by external gears, the sliding of their plungers past one another without contact is common to all rotational motion piston pumps. As a result, these pumps are well suited for delivering non-lubricating and abrasive liquids. This aspect fundamentally distinguishes this type of pump from gear pumps. The types are classified according to the number of impellers and the type of hub design (Fig. 10.59).

In some cases, the design selected for the displacer ranges from a multi-impeller to a gear type. However, the rotational motion piston pump is not a gear pump

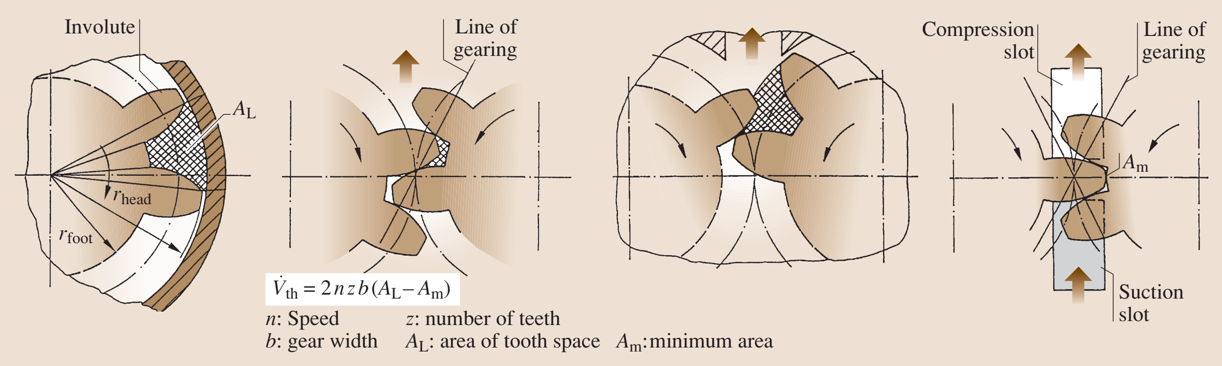



**Fig. 10.57** Displacement in a gear pump

since exterior synchronizer gears enable the displacer to run without contact.

The fixed hub produces a seal covering the space between the displacers. The plunger shape creates a flat seal in the space between the housing and the displacer so that the fixed hub is able to achieve better inner impermeability.

These pumps are used, for instance, to deliver finishes, paints, oils, gasoline, food and alcohol and for foaming liquid–gas mixtures.

The pressure range of these pumps reaches 20 bar at delivery rates up to 500 $m^3/h$. They can pump media with very high viscosities.

Other displacer forms, some without dead space, are also used for special applications such as those in the food processing industry (Fig. 10.60).

**Rotary Screw Pumps.** Among other things, rotary screw pumps are called *degenerate* gear pumps with helical gears and few teeth. The inter-rotating profiles effect an axial delivery. Rotary screw pumps are constructed with two, three or five spindles. The design with two or three spindles is most widespread. The secondary spindles are arranged around the main spindle (Fig. 10.61).

These pumps are engineered both with single and dual flow to compensate for the axial thrust. The main spindle is usually double-threaded; the secondary spindles double or triple threaded. Depending on the pumped medium, pitch and profile, they are constructed with or without synchronizer gears.

Basically three profiles are used:

- Involute profile, without synchronizer gears (e.g., Leistritz)
- Epicycloidal profile (e.g., IMO AB, Stockholm, Sweden: manufacturer of rotary positive 3 screw pumps)
- Trapezoidal or rectangular profile, only with synchronizer gears (e.g., Bornemann)

Rotary screw pumps are constructed for:

- Delivery rates of 10–150 $m^3/h$
- Pressures of 15–150 bar
- Speeds of 500–1500 $min^{-1}$
- Viscosities of $25 \times 10^{-6}$–$5 \times 10^{-3}$ $m^2/s$

**Planetary Rotation Piston Machine.**
*Eccentric Screw Pumps.* The principle of the eccentric screw pump (Fig. 10.62) goes back to the Frenchman Moineau. A rotor with a single-threaded diamond knurl rotates in a housing (stator) with a double-threaded diamond knurl. An alternative name for the eccentric screw

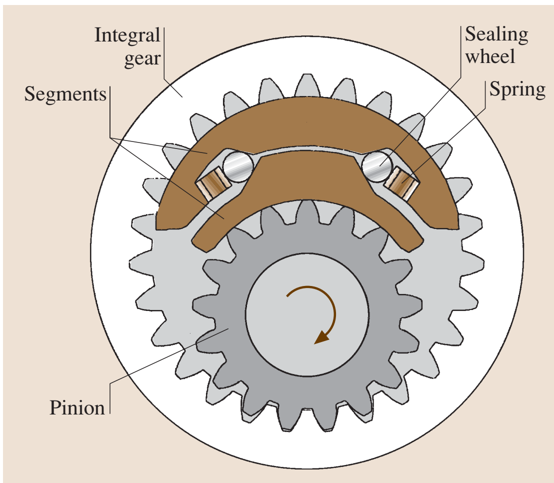



**Fig. 10.58** Internal gear pump with clearance compensation (after [10.15])

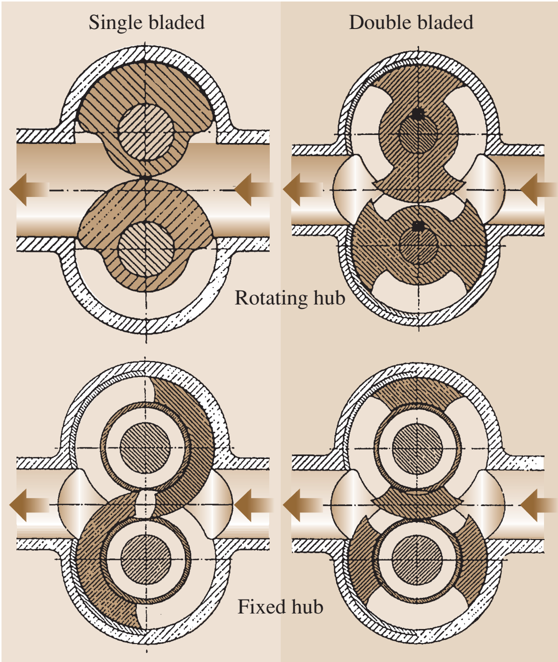



**Fig. 10.59** Variants of rotational motion piston pumps (after [10.16])

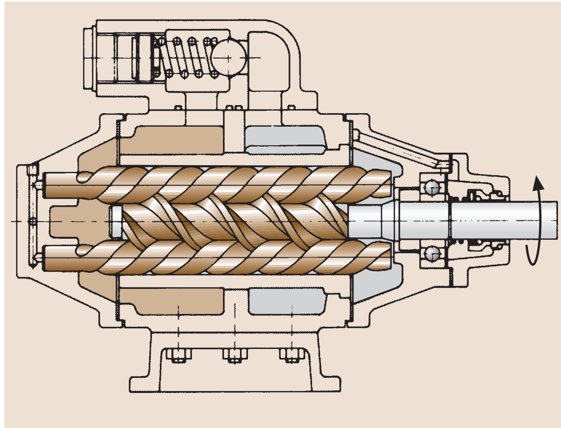


**Fig. 10.61** Rotary screw pumps with three spindles (after [10.17])

pump is the single-spindle pump. Depending on its design, this pump is an eccentric rotational motion piston pump.

At every point, the rotor has a circular section with its center lying on the rotor axis on a helix. The housing has an oval section and twice the pitch of the rotor.

Eccentric screw pumps attain delivery rates of 0.6–1000 $m^3/h$ at pressures of up to 40 bar. Apart from being able to deliver abrasive and highly viscose media, the eccentric screw pump provides the advantage of a pulsation-free delivery rate (Fig. 10.63). Since dry running the pump has to be avoided in any operational situation, the suction and compression flanges in this type of pump are frequently arranged on the top side of the housing.

Typically, the rotors are made of CrNi steel or cast iron and the housing of red brass or cast iron. The stator is made of elastic material such as rubber, polytetrafluoroethylene (PTFE) or neoprene.

The medium may contain a solid content of up to 45%. However, this leads to a sharp power increase and

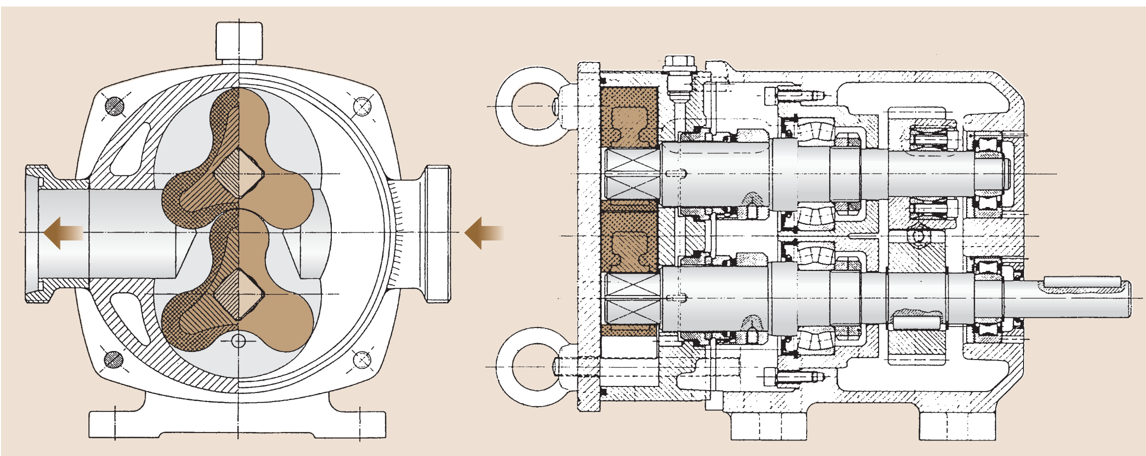


**Fig. 10.60** Rotational motion piston pump for food, cross and longitudinal sections (after [10.16])

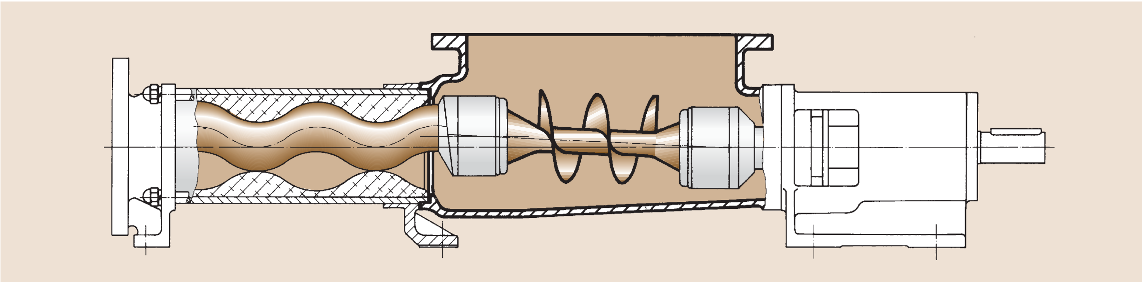


**Fig. 10.62** Eccentric screw pump (after [10.18])

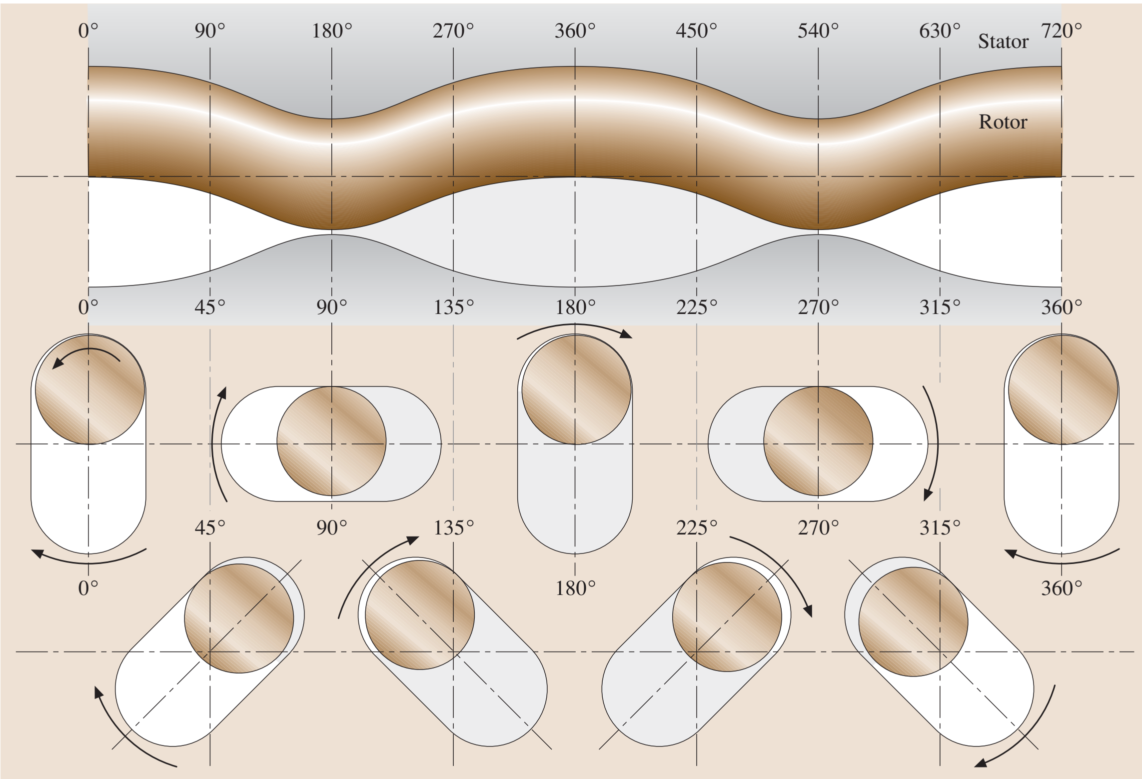



**Fig. 10.63** Displacement effect of the eccentric screw pump (after [10.18])

causes low speeds. The elastic stator makes it possible to attain high impermeability and a suction head of up to 8.5 m.

The rotor executes a circular motion with its center axle so that it must be driven by cardan joints. In some cases, only one cardan joint is used in smaller units. As a result, the stator must absorb part of the motion. In addition, a joint can be used, which compensates for a parallel axle offset. Impellers or screws that improve the feeding of the pumped substance are often mounted on the cardan shafts.

Eccentric screw pumps are used to pump mortar, among other things.

### Circulation Piston Machine.

*Cell Pumps.* Cell pumps (Fig. 10.64) are frequently used for low delivery rates of up to 25 $m^3/h$. As a rule, four to six slide valves are employed. Up to 12 slide valves are used at higher pressures of up to 100 bar. In some cases,

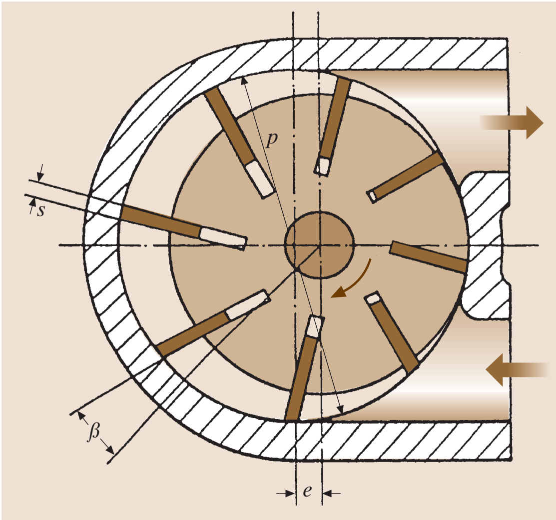



Fig. 10.64 Cell pump

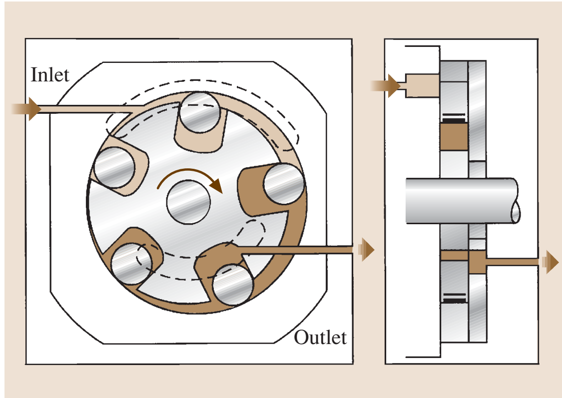



Fig. 10.65 Roller cell fuel pump (after [10.19])

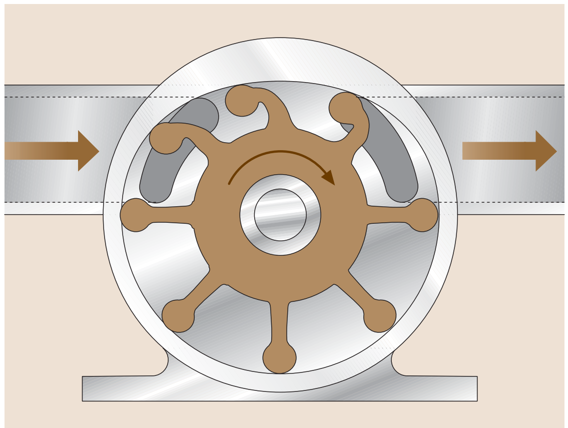


Fig. 10.66 Impeller pumps (after [10.20])

even two slide valves per channel are used to increase impermeability.

Normally, the centrifugal force alone lifts the slide valves negatively. As a result, these pumps are relatively leaky at low speeds. The principle is broadly applied to fuel feed pumps for instance. The roller cell pump is another common type. Rollers are used instead of slide valves. Their shape gives them advantages over slide valves in terms of wear.

This type of pump is used for example as an electric fuel pump to pump gasoline in cars (Fig. 10.65).

So-called impeller pumps are also used when delivery flows are smaller, e.g., for garden pumps or coolant pumps for boat engines (Fig. 10.66).

The working chamber changes volume by bending the rotor's elastic, usually rubber impellers against the sickle-shaped part of the housing deviating from a circular shape.

The rotor is installed axially without any gap in order to guarantee high impermeability. Thus, these pumps are self-priming. The impellers' ability to upright themselves determines the maximum speed. The delivery pressure depends on the flexural strength of the impellers. These pumps are insensitive to contamination of the pumped medium but can only reach low pressures of approximately 4 bar. They are used as boat motor seawater pumps and as garden pumps.

## 10.3 Compressors

Compressors can be divided into two general categories: positive displacement type and non-positive displacement types. Positive displacement compressors deliver essentially the same volume of air regardless of the pressure ratio while non-positive displacement compressors will have reduced flow at higher delivery pressures. Axial flow and centrifugal compressors are non-positive displacement types and reciprocating, rotary-screw, and diaphragm compressors are of the positive displacement type. This section discusses the characteristics of positive displacement reciprocating compressors.

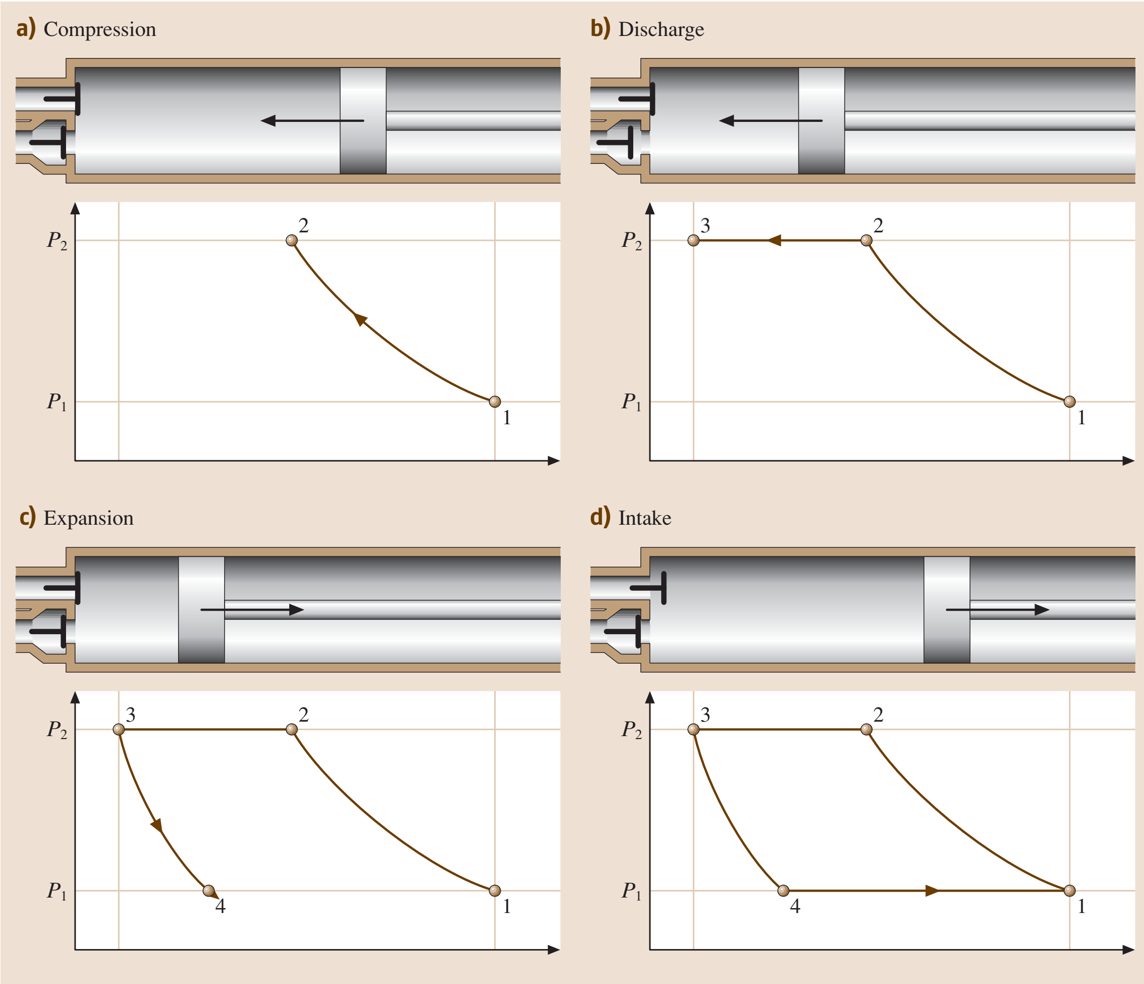



**Fig. 10.67a–d** Reciprocating compression processes: **(a)** compression **(b)** discharge **(c)** expansion **(d)** intake

Reciprocating compressors are frequently used when high pressures are needed, although they generally use multiple stages of compression to limit the power required for the process. The basic compression process can be visualized as occurring in a piston–cylinder configuration that is similar to that of an internal combustion engine. In practice, reciprocating compressors frequently use both sides of the piston in *double-acting* systems. Inlet and discharge valves are spring-loaded so that they will automatically open when there is an appropriate pressure in the cylinder. The inlet valve will open when the cylinder pressure drops below the inlet pressure, and the discharge valve will open when the cylinder pressure exceeds the receiver pressure.

## 10.3.1 Cycle Description

Figure 10.67 shows the sequence of processes followed by reciprocating compressors. The upper part of each diagram shows the piston position in the cylinder and whether the valves are open or closed. The lower part of each diagram shows the corresponding pressure–volume diagram for the process. In Fig. 10.67a the piston moves to decrease the volume with the valves closed. The pressure rises from state 1 to state 2 following a process line that depends on the amount of heat loss during the compression process. When the cylinder pressure exceeds the receiver pressure, the discharge valve will be forced open and the gas is discharged at constant pressure, as shown in Fig. 10.67b. When the

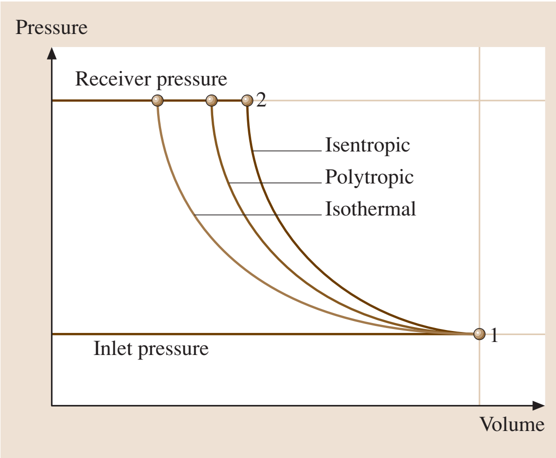



**Fig. 10.68** Compression process lines

piston reaches the point of minimum volume, the cylinder volume is equal to the clearance volume. This is the starting point for the expansion process from state 3 to state 4, shown in Fig. 10.67c, where the pressure falls from the receiver pressure to the intake pressure. Finally, when the cylinder pressure has dropped to the inlet pressure, the intake valve opens and fresh air is drawn into the cylinder as shown in Fig. 10.67d.

The compression process for a reciprocating compressor will generally follow a polytropic process line that is between a reversible adiabatic process (isentropic) and a constant temperature (isothermal) process as shown in Fig. 10.68. The isothermal process accomplishes the compression with the least work input. Ideal gases follow the relationship

$$pV = RT . \tag{10.62}$$

When an ideal gas is compressed it usually follows a polytropic process line that is modeled as

$$pV^n = \text{const} . \tag{10.63}$$

The usual values for $n$ are between 1, which corresponds to an isothermal process and $k$, which is the ratio of specific heats and is equal to 1.4 for air at 25 °C.

For the polytropic process, the pressure at the end of the compression process will depend on the compression ratio $r_v = \frac{V_1}{V_2}$ according to the following equation

$$p_2 = p_1 \left(\frac{V_1}{V_2}\right)^n = p_1 r_v^n . \tag{10.64}$$

The final temperature can be calculated as

$$T_2 = T_1 r_v^{n-1} = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} = T_1 r_p^{\frac{n-1}{n}} . \tag{10.65}$$

The power required for the compression process is

$$Pwr = p_1 \dot{v}_1 \left(\frac{n-1}{n}\right)\left(r_p^{\frac{n-1}{n}} - 1\right) , \tag{10.66}$$

where $\dot{v}_1$ is the volumetric flow rate entering the compressor, $r_p$ is the pressure ratio $p_2/p_1$, $n$ is the polytropic exponent, and $p_1$ and $p_2$ are the pressures at states 1 and 2, respectively.

At high pressures, many gases will deviate considerably from ideal gas behavior. Usually, these deviations are incorporated into the calculation through the use of a compressibility factor $Z$. The equation between temperature, pressure and volume becomes

$$pV = ZRT . \tag{10.67}$$

The value of $Z$ depends on the state, and charts are widely available for most common gases.

While reciprocating compressors are usually designed to encourage heat rejection, it is more common to have an intercooler between separate stages of compression. The intercoolers and multiple compressor stages are usually integrated into a common assembly that may take advantage of double-acting pistons.

### 10.3.2 Multi-Staging

Compression at constant temperature requires less work input than when the temperature is allowed to rise. In practice, isothermal compression is difficult to achieve. However, incorporating intercooling between multiple stages of compression can approximate isothermal compression.

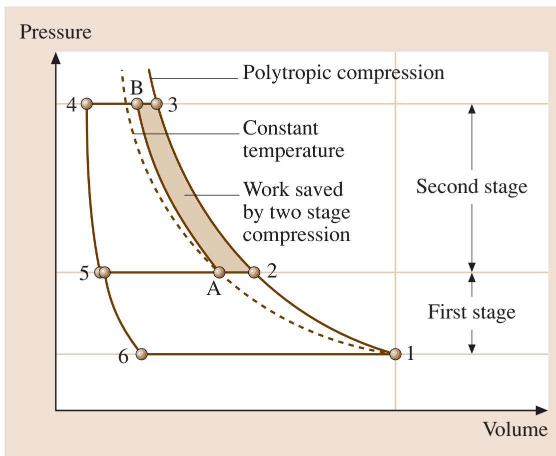



**Fig. 10.69** Two stage compression

In the limiting case of an infinite number of compression stages that include intercooling back to the inlet temperature, the total work required will be equal to that for isothermal compression. Even the division of a compression process into two stages can save considerable work input.

A two-stage compression process is depicted in Fig. 10.69. The compression process begins at state 1, which corresponds to the piston at BDC and inlet pressure. The gas is compressed in the first stage to the inter-cooler discharge pressure at state 2. The gas is then discharged at $p_2$ into an intercooler before entering the second compressor stage at state A. In this idealized case, the intercooler is assumed to operate at constant pressure but the volume of gas is decreased due to the drop in temperature to the initial temperature of the gas. Then, the gas is compressed to state B before being discharged at $p_\mathrm{B}$. If the pressure increase had been attempted with a single stage of compression, then the process line would pass through states 1, 2, and 3. Since the area enclosed on the $p$–$V$ diagram is the work needed to accomplish the process, the shaded area is the difference in work between the single stage and two-stage compression processes. Clearly, the two stage compression is more efficient.

The optimum pressure for intercooling is generally assumed to correspond to an equal pressure ratio for each stage. This assumes the intercooling is able to reduce the gas temperature to the inlet temperature at each stage. If $r_s$ is the pressure ratio for each stage, $r_t$ is the overall pressure ratio, and $s$ is the number of stages, then

$$r_s = \sqrt[s]{r_t}\,. \qquad (10.68)$$

### 10.3.3 Design Factors

At the end of the compressor's discharge stroke, shown in Fig. 10.67b, gas fills the clearance volume at the discharge pressure. This gas expands as the piston moves away from the cylinder head until its pressure drops below the inlet pressure when the intake valve opens. The induction of fresh charge does not begin until this point is reached so the full volume displaced by the piston is not utilized. When the clearance volume is large, then the capacity of the compressor is less.

The volumetric efficiency of a compressor can be approximated as

$$\eta_v = 1 - \frac{V_\mathrm{clearance}}{V_\mathrm{Displ}}\left(r_\mathrm{p}^{\frac{1}{k}} - 1\right) - \mathrm{Leakage}\,, \qquad (10.69)$$

where the leakage can generally be assumed to be between 0.03 and 0.05. Lower-molecular-weight gases usually have higher leakage.

It can be seen from the equation for $\eta_\mathrm{v}$ that an increase in clearance volume directly causes a decrease in volumetric efficiency. Its significance is much greater for high values of the pressure ratio $r_\mathrm{p}$.

Although compressor designers try to minimize it, the clearance volume cannot be entirely eliminated. Reducing it to less than 4% of the displacement volume is difficult. The amount of clearance volume will affect the capacity of the compressor and its efficiency. Overall compression efficiency is improved when valve flow area is large. However, the desire to minimize the clearance volume conflicts with the desire to maintain large valves. Thus, there is often a tradeoff between volumetric efficiency and compression efficiency, which determines the actual value of the clearance volume.

## 10.4 Internal Combustion Engines

In an internal combustion engine, the working fluid consists of a fuel–air mixture and the combustion products of this mixture. Although many cycles have been proposed, the traditional two-stroke and four-stroke cycles still dominate current use.

Depending on their design and application, internal combustion engines provide excellent portability, power density, and fuel economy. Vehicles that utilize internal combustion engines provide unsurpassed range, drivability, and driver comfort while maintaining low levels of hazardous pollutants.

### 10.4.1 Basic Engine Types

Engines can be categorized in many different ways. The number of cylinders, the type of valve actuation, and whether the engine is turbocharged or naturally aspirated are all possible choices. Some engines are spark-ignited and utilize a homogeneous fuel–air mixture and some are compression-ignited, also called diesel engines, and utilize a heterogeneous fuel–air mixture. Another characteristic is whether the engine uses the two-stroke or four-stroke engine cycle.

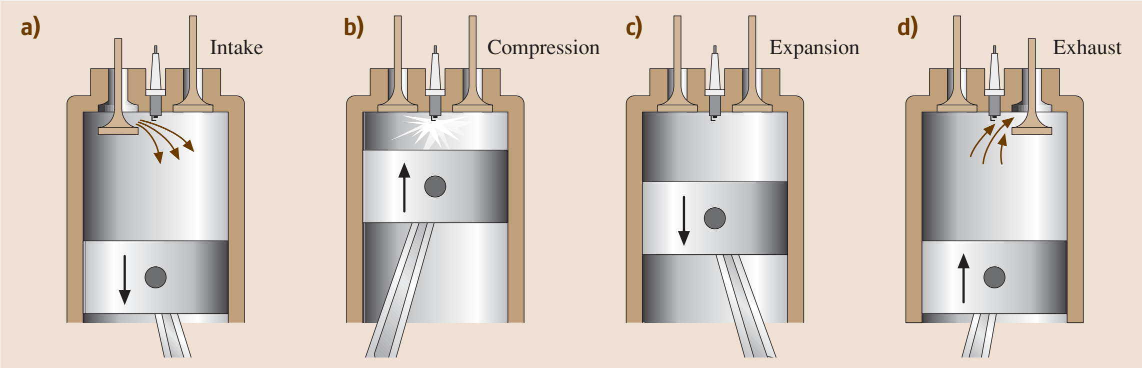



**Fig. 10.70a–d** Four stroke cycle events

All reciprocating internal combustion engines need to go through the four processes of intake, compression, expansion, and exhaust. The basic difference between two-stroke and four-stroke cycle engines is that the two-stroke engine accomplishes the four processes in a single revolution, or two strokes of the piston (one up and one down). The four-stroke engine needs two revolutions to complete the cycle.

The processes are shown in Fig. 10.70 for a four stroke cycle. In Fig. 10.70a, the intake valve is open and fresh charge is drawn in as the piston moves downward toward the bottom dead center (BDC) position. After BDC, the intake valve closes and the piston compresses the air on its upward compression stroke. Near the end of the compression stroke, close to top dead center (TDC), the spark plug fires and ignites the fuel–air mixture. In a diesel engine, only air is drawn in through the intake valve and fuel is injected into the air near the end of the compression stroke. This fuel self-ignites after a short delay. For both spark-ignited and diesel engines, the combustion products do work on the piston as it moves out for the expansion stroke as shown in Fig. 10.70c. In Fig. 10.70d, the exhaust valve opens near BDC and the combustion products are expelled from the cylinder by the upward motion of the piston. At the end of the exhaust stroke, the intake valve opens and the cycle is repeated with another intake stroke.

The two-stroke cycle is depicted in Fig. 10.71. When the piston is near the BDC position, air enters the cylinder from a port in the cylinder wall. There is a deflector on the top of the piston to inhibit the direct passage of the fresh charge across the cylinder and out the exhaust port on the other side. As the piston moves upward, it covers the intake and exhaust ports and compresses the fuel–air mixture. Near TDC, the spark fires to start the combustion process. In a two-stroke diesel engine the fuel is injected into the compressed air at this point. For both types of engines, the combustion products expand and do work on the piston surface until the point where the piston uncovers the upper edge of the exhaust port. At this point, the gases in the cylinder rapidly blow down until the pressure in the cylinder

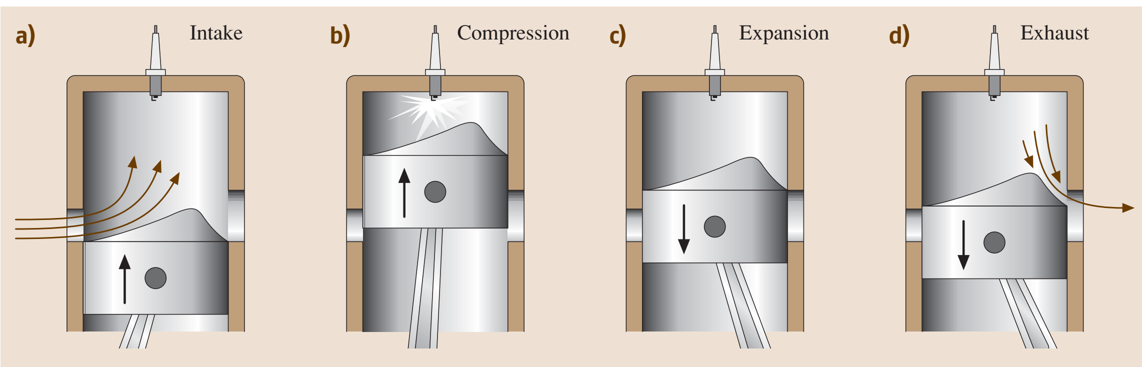



**Fig. 10.71a–d** Two stroke cycle events

is close to the exhaust pressure. As the piston continues its downward motion, it uncovers the intake port so fresh charge can enter the cylinder and repeat the cycle.

Because the two-stroke engine needs to complete all four processes in a single revolution, it must start the exhaust process well before the piston has reached bottom dead center. A four-stroke engine can wait until about 140° after TDC before starting to open the exhaust valve since the primary criterion is that the period of rapid pressure equalization that occurs when the valve is first opened, called the blowdown, is essentially complete by BDC. A two-stroke engine must start opening the exhaust port at about 90° after TDC to provide sufficient time for the blowdown before the intake port opens and the intake process starts. Opening the exhaust valve early so that the cylinder pressure is throttled down to ambient allows no recovery of the energy in those hot gases and is the major reason why two-stroke engines are less efficient than four-stroke engines.

In four-stroke engines, almost the entire cylinder contents of burned product gases, called residual gases, is expelled when the piston reaches TDC at the end of the exhaust stroke. Then, as the piston moves away from TDC, it produces a low pressure in the cylinder which draws in a fresh air charge through the intake valve. Four stroke engines are said to be self-scavenging. That is, the piston motion is directly responsible for moving the exhaust gases out of the engine and drawing in fresh air. Two stroke engines are not self-scavenging. Some mechanism other than piston motion is needed to exchange the gases in the cylinder. By opening the exhaust valve early, while the cylinder pressure is still high, most of the residual gases can be expelled. The fresh charge must be forced into the cylinder from a pressurized source, which might be a crankcase that is pressurized by the downward motion of the piston as in single cylinder two-stroke engines used for hand-held power equipment. It might also be from an engine-driven blower as is common in diesel two-stroke engines.

### 10.4.2 Performance Parameters

Engine speed and torque are the two most fundamental quantities of engine performance. Both are usually measured quantities with speed given in revolutions per minute (rpm) and torque in Newton-meters (N m). Power is defined as torque times rotational speed. When the torque is measured at the engine flywheel, it is called the *brake torque* and the power calculated from it is the brake power, as

$$P_\mathrm{b} = T_\mathrm{b}\omega = T_\mathrm{b} 2\pi N \,, \tag{10.70}$$

where $P_\mathrm{b}$ is the brake power, $T_\mathrm{b}$ is the brake torque, $\omega$ is the rotational speed, usually in radians/unit time, and $N$ is the rotational speed in rev/unit time.

The source of power in the engine is the work done on the piston by the expanding combustion gases. The power associated with this piston work is called the *indicated power*. The difference between the indicated power and the brake power is the power required to overcome friction and to drive the accessories including the water and oil pumps

$$P_\mathrm{i} = P_\mathrm{b} + P_\mathrm{f} \,, \tag{10.71}$$

where $P_\mathrm{i}$ is the indicated power, $P_\mathrm{b}$ is the brake power, and $P_\mathrm{f}$ is the friction power.

Direct calculation of the indicated power requires measurement of the cylinder pressure. The brake power can be calculated from the measured engine torque and speed. The friction power must be calculated from the difference of the indicated and brake power.

Generally, small engines run at high rotating speeds and large engines run at low rotating speeds. To allow comparisons between engines of different sizes, it is common to calculate the *mean piston speed*, which is the average velocity of the piston as the engine makes one revolution

$$\mathrm{MPS} = \frac{2S}{\text{time for one revolution}} = 2SN \,, \tag{10.72}$$

where $S$ is the stroke, and $N$ is the engine rotating speed.

The *mean effective pressure* (MEP) is a way to normalize the work done by the engine against the size of the engine. It is intended as a measure of engine loading. The MEP is defined as the ratio of the work done by the engine in one cycle to the displacement volume. A four-stroke engine undergoes one cycle in two revolutions (or $4\pi$ radians) so the work done is equal to the torque times the angular displacement. When the brake torque is used, the quantity is known as the *break mean effective pressure* (BMEP)

$$\mathrm{BMEP} = \frac{4\pi T_b}{V_d} \,. \tag{10.73}$$

Since work is measured in N m and volume in m$^3$, the ratio has units of pressure N/m$^2$. The MEP is sometimes described as the pressure which, if applied as a constant pressure during the expansion stroke, would give the same work as actually produced by the engine.

The *mechanical efficiency* is a measure of how much of the power produced by the combustion process is delivered to the output shaft. It is defined as the ratio of the brake power to the indicated power

$$\eta_m = \frac{P_b}{P_i} = 1 - \frac{P_f}{P_i}\,, \tag{10.74}$$

where $\eta_m$ is the mechanical efficiency, $P_b$ is the brake power, $P_i$ is the indicated power, and $P_f$ is the friction power.

The *thermal efficiency* is defined as the ratio of the power produced by the engine to the rate at which fuel energy is supplied to the engine, as indicated by the lower heating value (LHV). When the brake power is used, the quantity is known as the brake thermal efficiency

$$\text{Brake thermal efficiency} = \eta_{bt} = \frac{P_b}{\dot{m}_{fuel}\,(\text{LHV})}\,. \tag{10.75}$$

The mechanical and thermal efficiencies are sometimes confused. For a modern engine running at full load, the mechanical efficiency may be 90% or higher. However, the thermal efficiency will generally be 30–45%.

The *specific fuel consumption* (SFC) is the ratio of the fuel flow rate to the power of the engine. When brake power is used, the quantity is known as the brake specific fuel consumption

$$\text{BSFC} = \frac{\dot{m}_{fuel}}{P_b}\,. \tag{10.76}$$

The BSFC is similar to an efficiency in that it measures how little fuel may be required to do a certain quantity of work. The lower the BSFC, the more efficient the engine.

The *volumetric efficiency* is a measure of how well air moves through the engine. For a four-stroke engine

$$\eta_v = \frac{\dot{m}_{actual}}{\dot{m}_{ideal}} = \frac{\dot{m}_{actual}}{\rho_{ref} V_d \left(\frac{\text{rpm}}{2}\right)}\,, \tag{10.77}$$

where $\eta_v$ is the volumetric efficiency, $\dot{m}_{actual}$ is the actual mass flow rate of air (or air–fuel mixture) entering the engine, $\dot{m}_{ideal}$ is the ideal mass flow rate of air (or air–fuel mixture), $\rho_{ref}$ is a reference density, $V_d$ is the displacement volume, and $\frac{\text{rpm}}{2}$ is the number of engine cycles per minute.

For spark-ignited engines, the values of $\dot{m}_{actual}$ and $\dot{m}_{ideal}$ refer to the fuel–air mixture entering the engine. For diesel engines they refer only to the air entering the engine.

The volumetric efficiency tends to be ambiguous for several reasons:

1. There is uncertainty about where the reference density should be calculated. Some sources suggest using ambient conditions while others suggest using the average intake manifold pressure and temperature.
2. Although it is considered to be an efficiency, there is no reason why the volumetric efficiency cannot be greater than one. If the ambient density is used to compute the volumetric efficiency of a turbocharged engine, the volumetric efficiency may be as high as 2 or 3. Even a naturally aspirated engine with a tuned intake system can have a volumetric efficiency of 1.2 or 1.3.
3. In engines with a large valve overlap period, a significant amount of air can blow through the engine directly from the intake to the exhaust without participating in a combustion process. This air could contribute to a high volumetric efficiency but is not available for combustion.

## 10.4.3 Air Systems

The power produced by all internal combustion engines is limited by their ability to draw air from the atmosphere. Fuel systems can always be designed to provide the amount of fuel appropriate to this air flow. To increase the power produced by an engine of a certain displacement volume, the air flow needs to be increased. This can be done with passive techniques that are incorporated into the engine's design or through the addition of external devices, such as a supercharger.

### Natural Aspiration

In a naturally aspirated engine, the air flows into the cylinder through the intake manifold and intake ports without the use of an external compressor of blower. At high engine speed this air moves at high velocity. When the piston approaches the end of the intake stroke, the momentum of the air keeps the air moving toward the cylinder and can continue to force air into the cylinder after the piston has started upward on the compression stroke. By properly timing the closing of the intake valve, the amount of air trapped in the cylinder can be increased beyond that which would be predicted based on ambient air density. This phenomenon, called the *ram effect*, increases with air velocity and therefore with engine speed.

A second effect that can be utilized to increase engine air flow is to take advantage of the pressure waves that are induced in the intake and exhaust system due to valve opening and closing events and piston motion. For example, the high-pressure wave created when the exhaust valve opens and rapidly blows down the cylinder contents travels to the end of the exhaust pipe and is reflected as a low-pressure wave or rarefaction wave. If this wave is timed to enter the cylinder near the end of the exhaust stroke it can assist in evacuating the residual gases and draw in fresh charge as the intake valve opens. Similarly, rarefaction waves in the intake system are reflected from the open end of the intake as pressure waves that will force more air into the cylinder. This process, known as tuning, is highly dependent on the relationship between valve timing, pipe lengths, and the speed of sound in the intake and exhaust gases. As a result, the benefits of tuning tend to be concentrated at specific engine speeds and the effects at other speeds may actually be negative. Other passive effects involving resonator cavities connected to the intake and exhaust pipes can also be used to raise the air flow to the engine. The engine air flow can also be increased with the addition of a compressor in a technique known as supercharging to be discussed later.

### Effect of Speed on Volumetric Efficiency

Engine volumetric efficiency is affected by engine speed. At higher speeds, the pressure drop resulting from frictional effects associated with higher flow velocity tends to cause less air–fuel mixture to enter the cylinder. Up to a certain speed, this effect can be offset by the ram effect and tuned intake and exhaust pipes. These flow enhancing effects can keep the volumetric efficiency relatively high over most of the engine's operating range but at a certain speed, which depends mostly on valve area and mean piston speed, the volumetric efficiency drops off sharply.

### Poppet Valves

Flow through a poppet valve is usually modeled as the product of isentropic flow through a restricted area passage and a flow coefficient that varies with valve lift and geometry. This equation is valid through all flow ranges, except when the flow is choked

$$\dot{m} = C_{\mathrm{D}}\rho_0 c_0 A_x \times \sqrt{\left(\frac{2}{k-1}\right)\left[\left(\frac{P_x}{P_0}\right)^{\frac{2}{k}} - \left(\frac{P_x}{P_0}\right)^{\frac{k-1}{k}}\right]}, \quad (10.78)$$

where $\dot{m}$ is the actual mass flow rate through the valve, $C_{\mathrm{D}}$ is the discharge coefficient for the valve, $\rho_0$ is the density at the upstream stagnation state, $c_0$ is the speed of sound at the upstream stagnation state, $A_{\mathrm{x}}$ is the minimum flow area between the valve and seat, $k$ is the ratio of specific heats $\frac{C_{\mathrm{p}}}{C_{\mathrm{v}}}$, $P_{\mathrm{x}}$ is the downstream static pressure, and $P_0$ is the upstream stagnation pressure,

$C_{\mathrm{D}}$ is greatest at low valve lifts. Under these conditions the air flow fills the entire flow channel without the separations that reduce the effective flow area. As the valve opens further, the flow separates from the valve edge and the seat, which reduces the actual flow area and decreases $C_{\mathrm{D}}$. The value of $C_{\mathrm{D}}$ tends to be independent of the flow Reynolds number, except at low lift where $C_{\mathrm{D}}$ decreases as the Reynolds Number increases reflecting a greater significance of boundary layers on the flow.

### Valve Timing

The timing for the opening and closing of the intake and exhaust valves can have a major impact on the engine performance. In four-stroke engines, the exhaust valve opens about 120–140° after TDC. Earlier timing will reduce expansion work and later may delay the exhaust blowdown so that high cylinder pressures early in the exhaust stroke increase the pumping work. Intake valve closing typically occurs at 20–60° after BDC and has a strong effect on the engine's volumetric efficiency. Earlier closing may reduce the air flow into the cylinder due to the ram effect described earlier, especially at high engine speeds. Later closing delays the start of the compression process and may allow some of the fresh charge to backflow into the intake manifold. The timings for closing the exhaust valve and opening the intake valve are not as critical for engine performance.

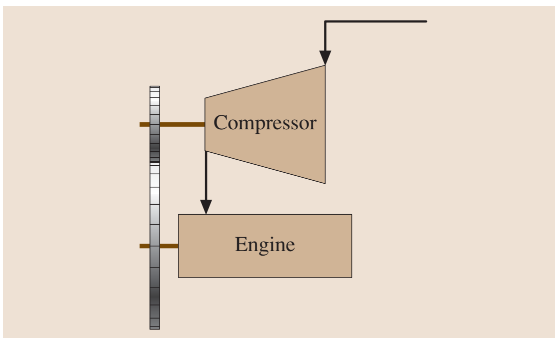



**Fig. 10.72** Engine-driven supercharger

### Supercharging

*Supercharging* is a general term for a variety of techniques used to boost the pressure of the air entering the cylinder to increase the engine's power. In some cases this involves devices to compress the air that might be driven directly by the engine or by the engine's exhaust gas. These devices are known as superchargers. When the device is driven by exhaust gas, it is called a turbo-supercharger or just turbocharger.

Figure 10.72 shows a schematic of an engine-driven supercharger that is driven by a set of gears, although belts are also frequently used. Engine-driven superchargers have the advantage that they provide air flow even at low speeds so they can be used to provide the scavenging needed for two-stroke engines during start-up. They also do not have the acceleration lag often noted with exhaust-driven turbochargers.

### Turbocharging

A specific type of supercharging that utilizes power recovered from expanding the exhaust gas to atmospheric pressure is called *turbocharging*. Figure 10.73 shows a schematic of a typical system. Air, after being filtered, is compressed and then supplied to the engine. Single-stage radial-design compressors are most common and are generally capable of 3 : 1 pressure ratios. The compressor is directly coupled to an expansion turbine that provides the work to drive the compressor. While the turbine imposes a back pressure on the engine, this is more than offset by the higher potential power available from the increased pressure in the intake system.

In diesel engines, the air supplied by the turbocharger is more dense and allows more fuel to be injected while still maintaining the air–fuel ratio limits imposed by exhaust emissions concerns. In fact, due to the greater availability of air, turbocharged engines can usually be operated at higher air–fuel ratios than naturally aspirated engines which improves both particulate and $NO_x$ emissions. Although engine power can be increased by a factor of 2–3 over a similarly sized naturally aspirated engine, fuel economy improves only slightly. This improvement is generally associated with the fact that friction losses do not increase proportionately with the increase in overall power so the mechanical efficiency of the engine improves. Turbocharging offers additional benefits of improved scavenging and piston cooling by allowing a portion of the compressed charge to short circuit through the cylinder during the valve overlap period.

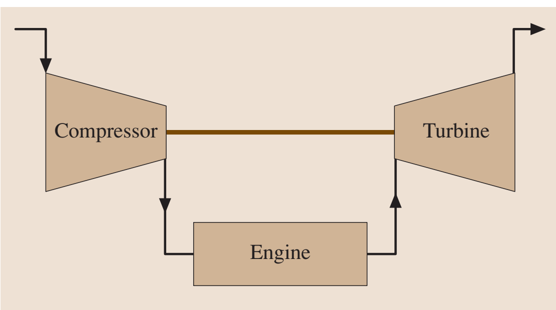



**Fig. 10.73** Typical turbocharger configuration

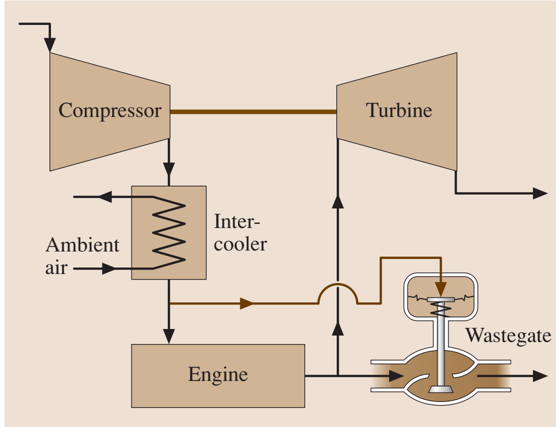



**Fig. 10.74** Turbocharged engine with intercooler and waste gate

Part of the density increase potential of the turbocharger is lessened by the fact that the air leaving the compressor is at high temperature. It is common to insert a heat exchanger, called an intercooler or aftercooler, into the airstream after the compressor. Figure 10.74 shows this configuration. The compressed air may be cooled by engine coolant, or more commonly in heavy-duty applications, by ambient air.

Also shown in Fig. 10.74 is a waste gate. This device allows a portion of the engine's exhaust gas to bypass the turbine. A diaphragm actuator senses compressor boost pressure and releases a portion of the exhaust gas so that the boost pressure does not become excessive. Turbocharger design and matching to a specific engine is usually a compromise between providing sufficient air at the low-speed peak-torque condition while not delivering excessive pressure at the high-speed high-load conditions that will cause high peak cylinder pressures.

For engines that are consistently operated at high loads, there is more energy available in the exhaust gas than is needed to operate the compressor. One option

to utilize this excess energy is to add a second turbine, sometimes known as the power turbine, that will recover energy from the exhaust gas leaving the first turbine, which drives the compressor. As shown in Fig. 10.75, the power turbine is connected to the engine's drive train through a mechanical connection so the power can be delivered to the engine's output shaft. This technique is often referred to as *turbocompounding*. The fuel economy benefits of turbocompounding can be significant, especially when the engine is designed for minimum heat rejection, which tends to increase exhaust energy. However, at current fuel prices, the savings in operational cost has not been enough to justify widespread acceptance of this technology.

Other technologies have been developed to provide optimum turbocharger performance over a larger fraction of the engine's operating range. Fig. 10.76 shows a twin-turbocharger technology developed by Opel that utilizes two turbochargers. As shown in the figure, exhaust gas from the engine is directed through two separate passage-ways to two turbochargers. The passage to the larger turbocharger is equipped with a flapper valve to limit the flow. At light load conditions, the flapper is mostly closed, which forces most of the exhaust flow through the small turbocharger so it has sufficient flow to operate at its most efficient condition. The large turbocharger is essentially free-spinning at this point and not providing much compression. As the engine speed and load increase, the exhaust flapper is opened allowing the larger turbocharger to become active so it can supply air for the high power condition. A check valve is provided in the intake pipe so that the high air flow does not all have to pass through the small turbocharger compressor. This arrangement provides efficient operation over a wide range of engine operating conditions.

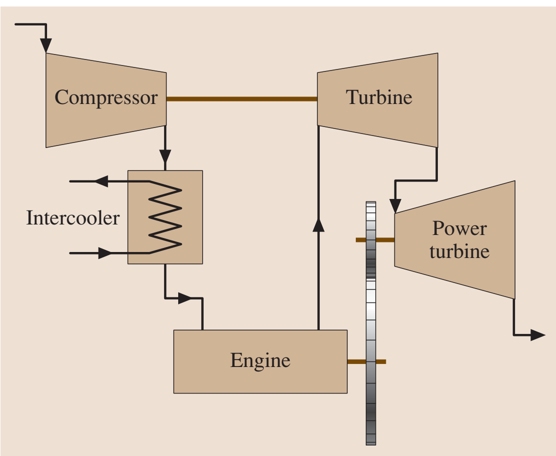



**Fig. 10.75** A turbocompounded engine

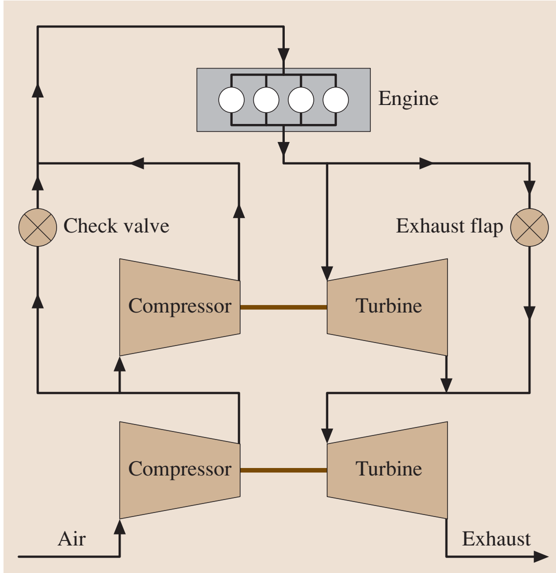



**Fig. 10.76** Twin turbocharger configuration

### Efficiency Definitions

The efficiency of a turbocharger is determined by the efficiency of its various elements. The efficiency of the compressor is calculated from the ratio of the work that would be required for a reversible adiabatic (isen-

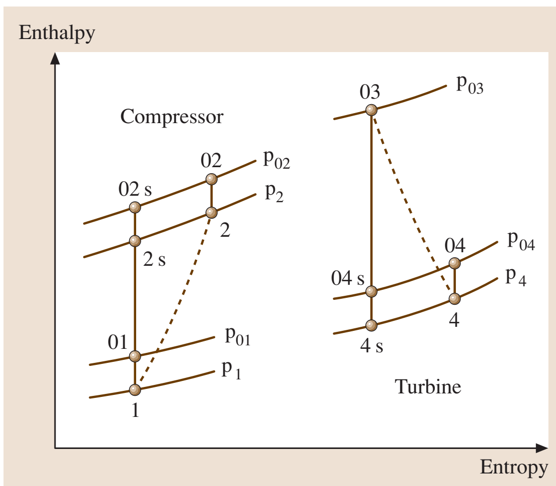



**Fig. 10.77** State definitions for compressor and turbine efficiencies

tropic) process divided by the actual work for the process. The compressor efficiency can be calculated using stagnation states at the inlet and outlet, or since the kinetic energy leaving the compressor is not recovered, the efficiency can be calculated from the inlet stagnation state to the outlet static state. The latter choice is more conservative and allows the effectiveness of the compressor outlet diffuser to be included in the efficiency. The equation for compressor efficiency is provided below based on the state definitions given in Fig. 10.77

$$\eta_C = \frac{h_{2s} - h_{01}}{h_{02} - h_{01}} . \tag{10.79}$$

In a similar manner, the efficiency of the turbine from the inlet stagnation state to the outlet static state can be written as

$$\eta_T = \frac{h_{03} - h_{04}}{h_{03} - h_{4s}} . \tag{10.80}$$

Although the simple design of the turbocharger offers little opportunity for frictional losses, a mechanical efficiency can be defined that consists of the power delivered to the compressor divided by the power produced by the turbine

$$\eta_M = \frac{-\dot{W}_C}{\dot{W}_T} . \tag{10.81}$$

### 10.4.4 Fuel Systems

This section will cover gasoline and diesel fuel systems. The principles, main designs, key operating characteristics and controls of each system will be explained. Other important adjuncts such as low-pressure systems, filtration and sensors will also be covered.

Gasoline systems are divided into carburation and fuel injection; fuel injection is further broken down into throttle body, port injection and direct injection.

Diesel systems are divided into cam-driven and common-rail systems; cam driven systems are further divided into two main groups: pump-line-nozzle (inline, distributor, unit pump) and unit injector.

#### Gasoline Fuel Systems

*Principle.* Gasoline fuel systems can be divided into two main types: carburation and fuel injection. For all systems the goal is to achieve a stoichiometric air–fuel ratio, which is the ideal ratio whereby all of the fuel is completely mixed and burned. For normal gasoline this is usually around 14.7 : 1 air mass to fuel mass.

*Carburation [10.21].* The simplest of all systems is the carburetor, which consists of the following subsystems:

- Inlet system to maintain a constant level of fuel in the reservoir
- Metering system to maintain the desired air–fuel ratio
- Accelerator-pump system to provide extra fuel during acceleration
- Power enrichment system to provide extra fuel during periods of high demand
- Choke system to provide a rich mixture for start and cold-engine operation

The carburetor uses the venturi principle: the inlet air flows through a necked-down area (venturi), where the flow increases in speed and decreases in pressure. A passage connects the fuel reservoir to the venturi; since the fuel in the reservoir is at atmospheric pressure, fuel flows from the reservoir to the lower-pressure area inside the venturi and then into the engine.

The pressure drop at the venturi increases with engine speed and with throttle position, thus causing fuel flow from the reservoir to the venturi to increase as engine speed and throttle position increase.

*Fuel Injection [10.22].* Fuel injection can be divided into two basic types: manifold (throttle body and port) and gasoline direct injection (GDI). Manifold injection sys-

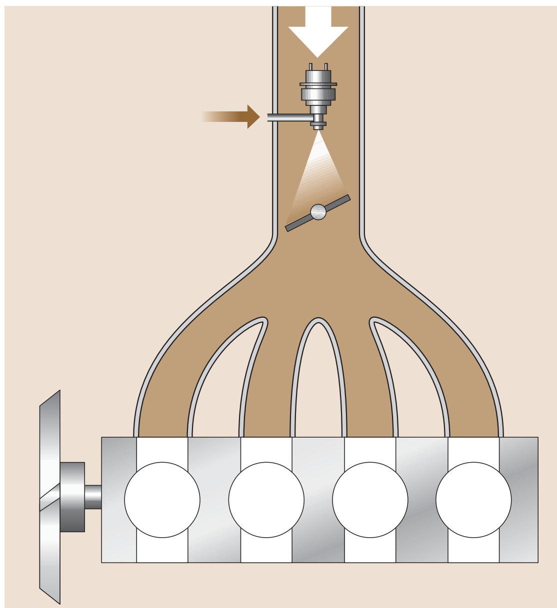


**Fig. 10.78** Throttle body injection

tems allow only a homogeneous operating mode; GDI allows this and several other modes as well.

*Throttle Body Injection.* Throttle body fuel injection is also known as single point because there is a central point of injection: a single electromagnetically operated injector is located directly above the throttle valve Fig. 10.78.

*Port Injection.* Port injection is also known as multipoint because fuel is injected into every intake port, i. e., onto the cylinder's intake valve Fig. 10.79.

There are four types of port injection:

- Simultaneous fuel injection: all injectors open and close together. Half of the fuel quantity is injected in one engine revolution; the remaining half is injected in the next revolution.
- Group fuel injection: the injectors are combined into two groups. All injectors in a group open and close together. One injector group injects the total fuel quantity needed for its cylinders in one engine revolution, then the second set injects its total fuel quantity in the next revolution.
- Sequential fuel injection (SEFI): fuel is injected individually for each cylinder. Injectors are triggered is the same sequence as the firing order. Duration and start of injection (relative to each cylinder's top dead center) are the same for all cylinders.
- Cylinder-individual fuel injection (CIFI): the duration of injection (i. e., fuel quantity) can be varied for each individual cylinder.

*Gasoline Direct Injection.* Fuel is injected directly into each cylinder's combustion chamber Fig. 10.80.

An electric fuel pump delivers fuel to the high-pressure pump, which pressurizes the fuel to 50–120 bar (depending upon the engine operating condition) and sent on to the accumulator rail. Since all injectors are connected in parallel to the rail, they are all constantly pressurized and inject only when an electric signal is sent to each injector.

GDI allows not only homogeneous operation but also stratified charge, homogeneous stratified charge, homogeneous anti-knock and stratified-charge/catalyst heating.

*Fuel Injection Control System.* The injected fuel quantity is determined by the control system which consists of sensors that measure various input parameters, a processor to determine the optimum fuel quantity based upon the sensor input, and actuators which carry out the commands of the processor.

Examples of sensors: inlet air temperature, airflow, accelerator pedal position, throttle-valve angle, rail-pressure, Lambda, coolant temperature, etc. Examples of actuators: injectors, idle-air control valve, throttle valve, fuel-pressure regulator, etc.

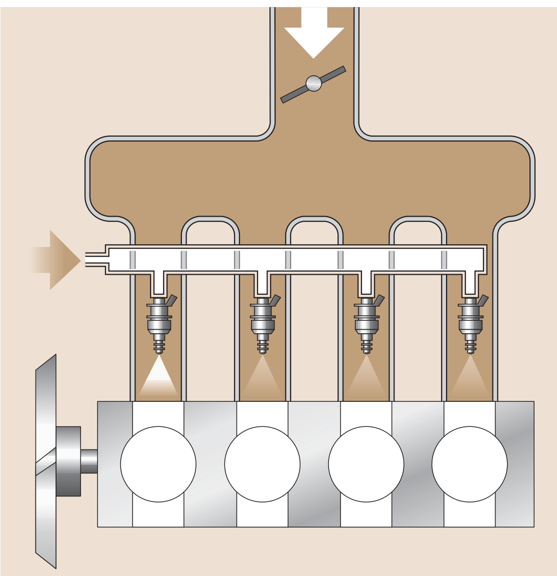


**Fig. 10.79** Port injection

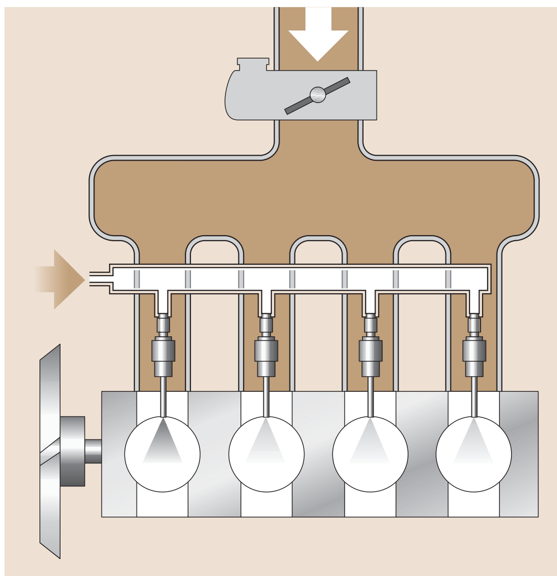


**Fig. 10.80** Gasoline direct injection

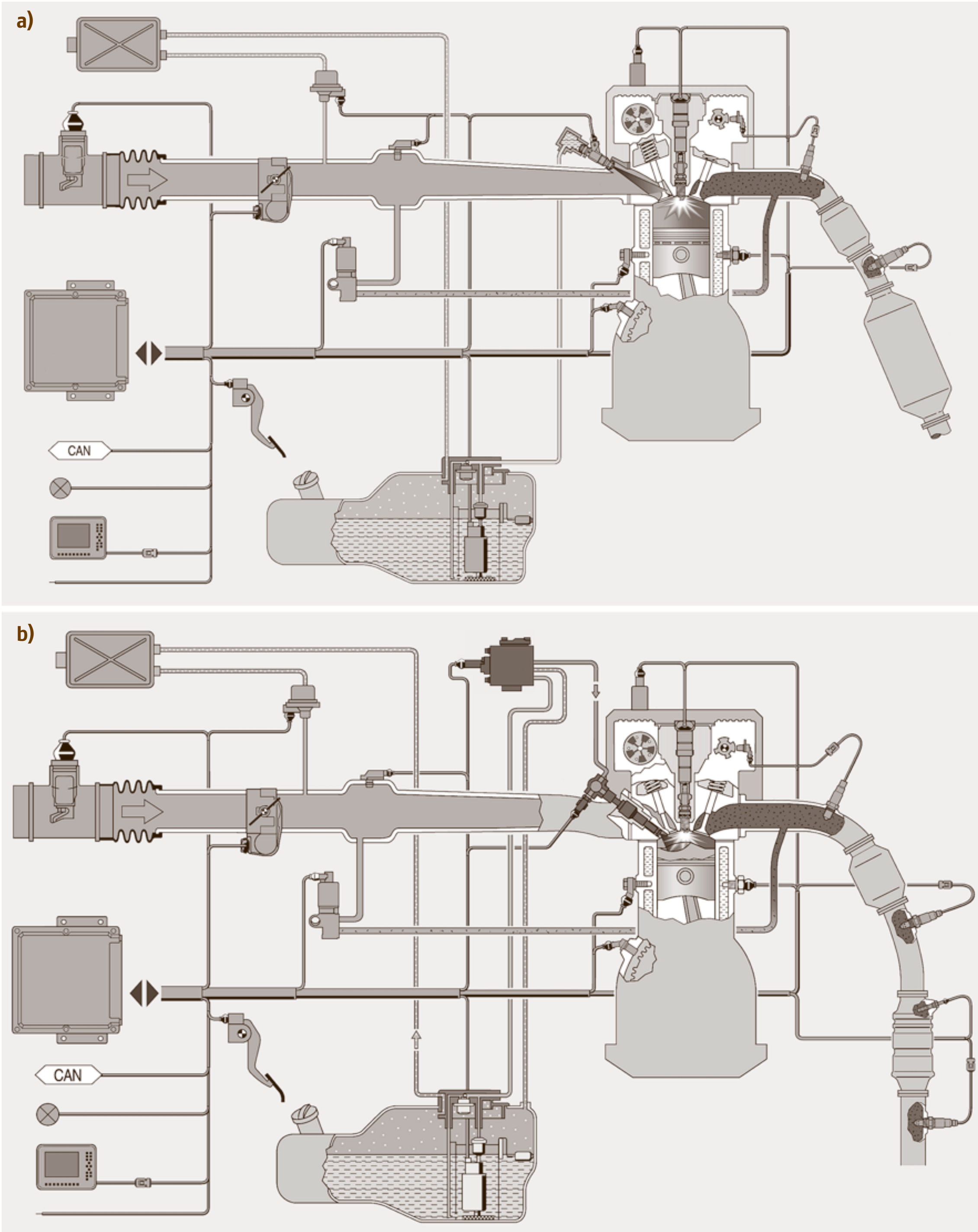



Fig. 10.81a,b Fuel injection control systems

The control system has to maintain the proper air–fuel ratio under the following engine operating modes:

- Start and warm-up
- Idle and part load
- Full load
- Acceleration and deceleration
- Overrun

The fuel injection control system may be combined with the ignition control system to allow for coordinated total engine management. The first image of Fig. 10.81 shows such a system for port injection and the second image of Fig. 10.81 shows a system for direct injection.

*Fuel Supply System.* For both carburetors and fuel injection, the fuel must be pumped from the storage tank by a mechanical or electrical pump, and must pass through a filter to remove impurities. The fuel pressure is regulated to a constant value.

### Diesel Fuel Injection Systems

*Principle.* There are essentially two types of diesel fuel injection: indirect and direct. Since the indirect diesel injection (IDI) is for pre-chamber or whirl-chamber engines, both of which are seldom being applied today, we will concentrate on direct injection (DI).

In the DI process the fuel is injected directly into the highly compressed hot air in the combustion chamber above the piston (Fig. 10.82). A multi-hole nozzle is used to distribute the fuel uniformly in the combustion chamber to ensure rapid mixing. Very high injection pressures (up to 2000 bar) are required to properly and completely atomize the fuel.

For any combination of engine operating parameters, the fuel injection system must deliver the correct amount of fuel, at the correct time, at the correct injection pressure, with the correct timing pattern, and at the correct point in each cylinder's combustion chamber. Limits such as emissions, combustion pressure, exhaust temperature, engine speed and torque and vehicle-specific loads may need to be taken into account when determining the proper fuel injection.

*Injection Characteristics.* The key parameters and their corresponding units for every fuel injection system are:

- Injected fuel quantity ($mm^3$/stroke or mg/stroke)
- Injection pressure (bar or psi)

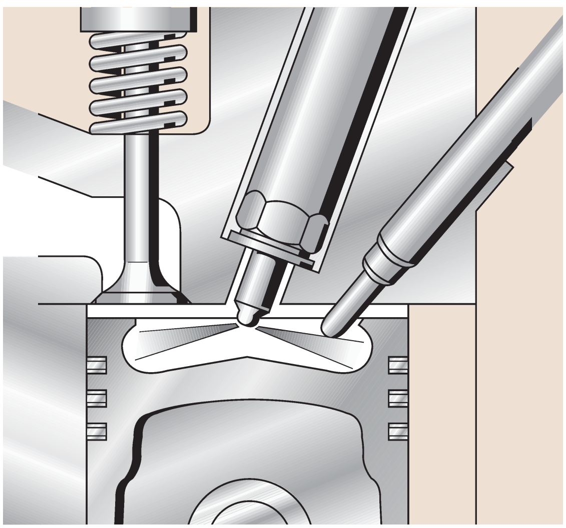


**Fig. 10.82** Direct injection process

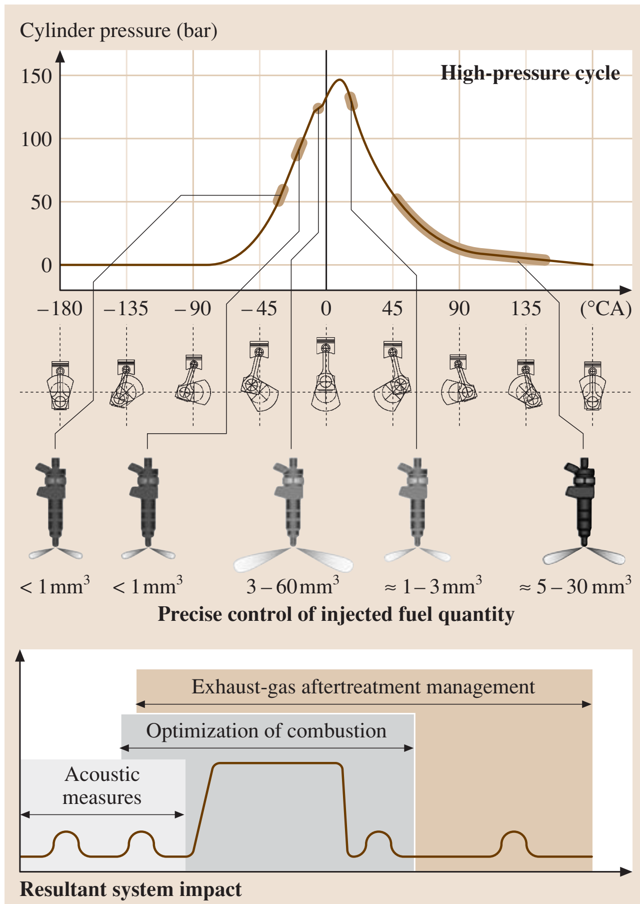



**Fig. 10.83** Injection pattern

- Injection duration (degrees crank angle)
- Injection timing (degrees before or after the engine piston's TDC)
- Injection spray plume:
  - Number of plumes
  - The plume angle (degrees)
  - Location of the plume in the combustion chamber (height above piston bowl, position in cylinder)
  - The shape of the plume itself

Depending upon the type of fuel injection system, one or more additional injection characteristics may be available:

- Injection rate shape ($mm^3$ as a function of crank or cam angle)
- Injection pattern (number and form of injections during each combustion cycle, Fig. 10.83; up to five injections per combustion cycle may be required)
- Pre-injection 1: to reduce noise, improve warm-up (by avoiding misfire and white smoke)
- Pre-injection 2: to further reduce noise, improve warm-up (by avoiding misfire and white smoke)
- Main injection
- Post-injection 1 (close after the main injection): to reduce soot emissions
- Post-injection 2 (retarded): to act as a reducing agent for an after-treatment device

*Fuel Injection System Designs [10.23].* To cover the wide variety of diesel engines in the marketplace (from motorcycles with 10 kW/cylinder up to railway locomotives with up to 1000 kW/cylinder), there is a corresponding variety of fuel injection designs. Figure 10.84 shows the various designs and Fig. 10.85 shows their corresponding injection pressure ranges. These can be divided into two main groups: cam driven and common rail.

*Cam-Driven Systems.* The cam-driven pumps can be further divided into two main groups: pump-line-nozzle and unit injector.

*Pumping Arrangement of Cam-Driven Systems.* The different designs within the pump-line-nozzle group:

- Inline fuel-injection pumps: one pumping unit per engine cylinder, mounted in a common housing
- Rotary fuel-injection pumps (axial-piston and radial-piston): one pumping unit for all engine cylinders; the fuel is distributed to each cylinder by a rotating shaft
- Individual-cylinder pumps: one pumping unit per engine cylinder, mounted separately and usually actuated by separate cam lobes on a common camshaft (Fig. 10.86)

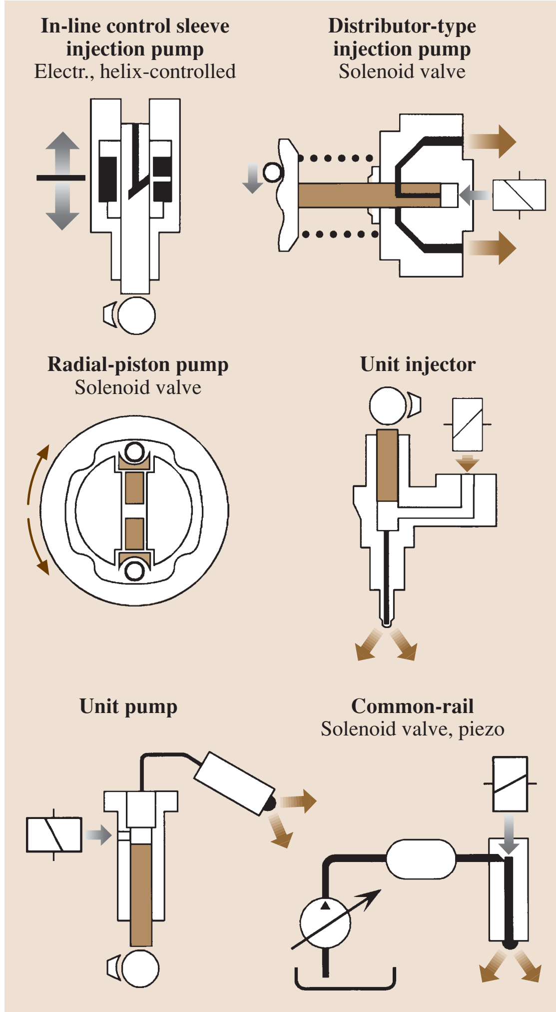



**Fig. 10.84** Fuel injection systems

Each of the above has a high-pressure line to connect each pumping mechanism to its corresponding nozzle.

The unit injector combines the pumping unit and nozzle in one assembly for each engine cylinder; each unit injector is actuated by a separate cam lobe on a common camshaft (Fig. 10.87).

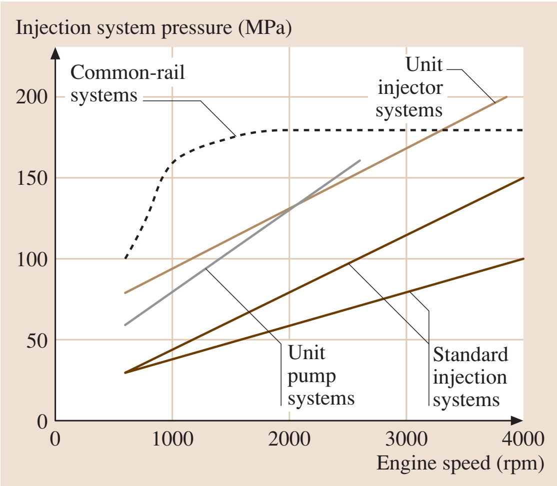



**Fig. 10.85** Pressure ranges of fuel injection systems

*Pumping Principle of Cam-Driven Systems.* The pumping principle is essentially the same for all cam-driven systems: the cam lobe, which is driven by the engine's crankshaft and is thus phased to the crankshaft, pushes a plunger which pressurizes the fuel. This pressurized fuel travels to the nozzle and as soon as the pressure exceeds the nozzle's opening pressure, the nozzle needle lifts and fuel is injected into the engine.

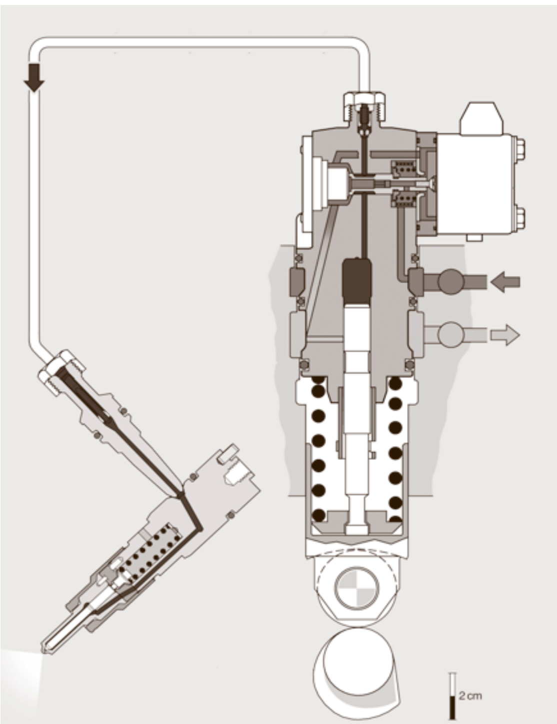



**Fig. 10.86** Individual-cylinder pump

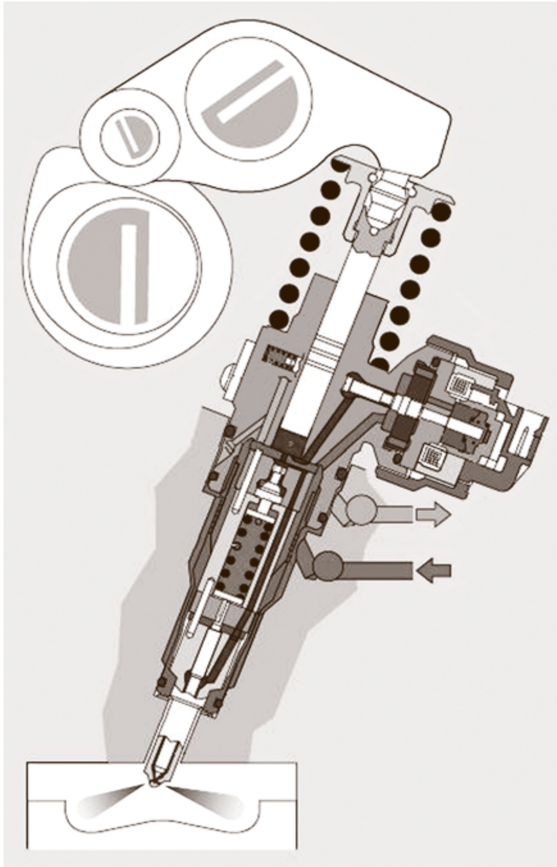


**Fig. 10.87** Unit injector

*Control System of Cam-Driven Systems.* The fuel quantity is controlled by varying the length of delivery, which is accomplished by varying either the beginning of delivery (and keeping the end constant), the end of delivery (keeping the beginning of delivery constant), or by varying both. Governing can be mechanical, pneumatic, electromechanical, or electronic, whereby the majority of new applications are electronic.

In addition to fuel control the timing may also be controllable, either by changing the phasing of the pumping cam to the crankshaft or by changing the point at which pressure starts to build up on the pumping cam.

Injection-rate and pilot injection may also be controlled in some instances.

*Common-Rail Systems.* This system offers the greatest flexibility in the choice of fuel-injection parameters.

*Pumping Arrangement.* The common-rail (CR) system utilizes a single pump to pressurize the fuel which is delivered to an accumulator rail. One injector per engine cylinder is connected to this rail by means of a high-pressure line (Fig. 10.88).

*Pumping Principle.* Fuel is delivered by the low-pressure system to the high-pressure pump, where it is pressurized by the pumping plungers (arranged either radially or inline) and sent from there to the accumulator rail. Since all injectors are connected in parallel to the rail, they are all constantly pressurized and inject only when an electric signal is sent to each injector.

*Control System.* The start of injection (timing) and duration of injection (fueling) is controlled by a solenoid valve (electromagnetic or piezo) on each injector. When the actuating signal is sent to the injector, a coil or piezo stack inside the injector is energized, upsetting the pressure balance (on both sides of the nozzle needle) that was holding the nozzle closed. The needle then lifts and the injector delivers fuel to the engine. When the signal ceases, the pressure above the needle increases, forcing the needle closed and thus ending injection.

In addition, a sensor in the high-pressure circuit monitors the system pressure and sends this information to the electronic control unit (ECU) so that the pressure can be regulated to a value that is optimal for the engine's operating condition.

*Low-Pressure System.* Regardless of the fuel injection design, all fuel injection systems require a primary fuel pump to deliver the fuel from the fuel tank through the fuel filter to the injection pump.

*Primary Fuel Pump.* The primary fuel pump can be either mechanical or electric; the electric pump can be inside the fuel tank, mounted on the engine, or mounted on the vehicle.

*Fuel Filter.* The fuel filter must strain out impurities in the fuel, as contamination will cause wear, orifice plugging, component sticking, and seizures. The filter medium must be fine enough to trap small particles and the filter size must be large enough to assure adequate service life. Often a preliminary filter is used in addition to the main filter to extend the filter change interval.

In addition, filters should have the ability to separate water out of the fuel, as water will cause wear, corrosion, and seizures.

Many filters today have water separation, a water-in-fuel indicator, and a heater combined in one unit.

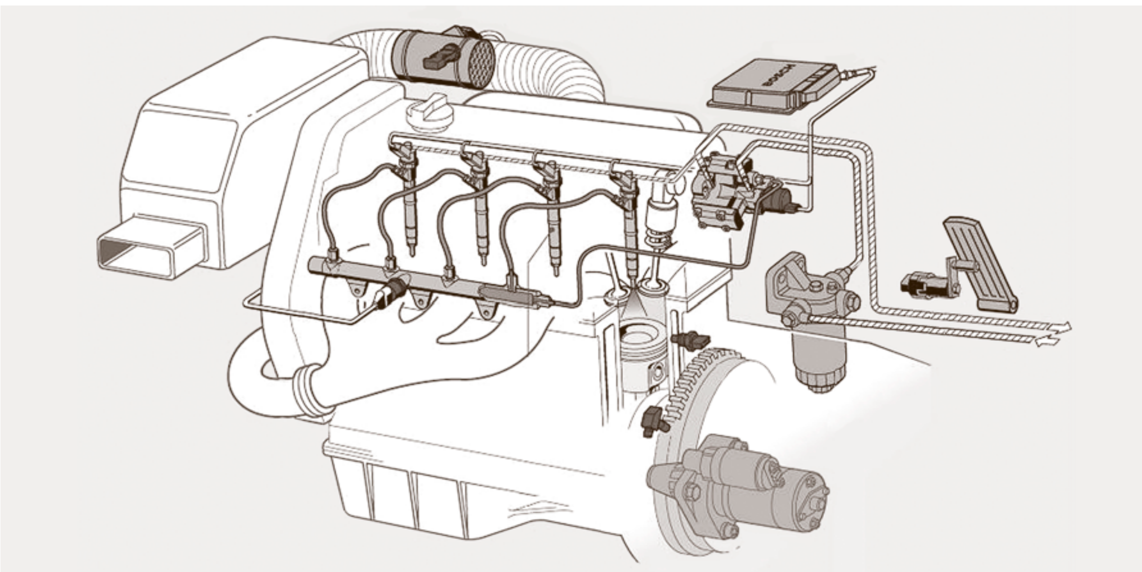


**Fig. 10.88** Common-rail system

## 10.4.5 Ignition Systems

This section will cover the principles of ignition and the two main design types. It will explain how the high voltage needed for ignition is generated and the importance of ignition timing. Lastly it will cover spark plug design and function.

There are two basic types of ignition system: conventional coil and electronic. The conventional coil design can have one of three types of trigger; the electronic design can either have a distributor or be distributor-less. All designs use essentially the same method for generating high voltage, and all systems use the same design type of spark plug.

### Principle

A gasoline internal combustion engine needs a spark to ignite the compressed air–fuel mixture in the combustion chamber. The spark is a discharge between the two electrodes that protrude into the combustion chamber. The ignition system generates the high voltage (up to 30 000 V) [10.24] needed to create the spark discharge and also initiates the spark to occur at the proper piston position (ignition timing).

### Ignition System Design

*Conventional Coil Ignition.* This system consists of an ignition coil, ignition distributor, and spark plugs; see Fig. 10.89. As the coil is similar for all types of systems it is explained under point 3 (*high-voltage generation*).

The distributor rotates in sequence with the engine's crankshaft and at half of the crankshaft's speed for four-stroke engines or at crankshaft speed for two-stroke engines. One of three types of triggers is used in the distributor to control the current through the ignition coil:

- Mechanical breaker points: a mechanical switch that is closed and opened once per firing event by a cam located on the distributor shaft. The number of cam lobes equals the numbers of cylinders.
- Breaker-triggered transistorized ignition: this design is similar to mechanical breaker points except the primary ignition circuit is controlled by a transistor instead of by the breaker points. Only the control current is switched by the breaker points; this extends the breaker-point life and allows higher primary currents to be controlled.
- Transistorized ignition with Hall-effect trigger or induction-type pulse generator: the breaker points are totally eliminated and replaced by either a Hall-effect sensor or an induction-type pulse generator located in the distributor. The sensor or generator create one signal per cylinder; this signal is used to charge and discharge the coil.

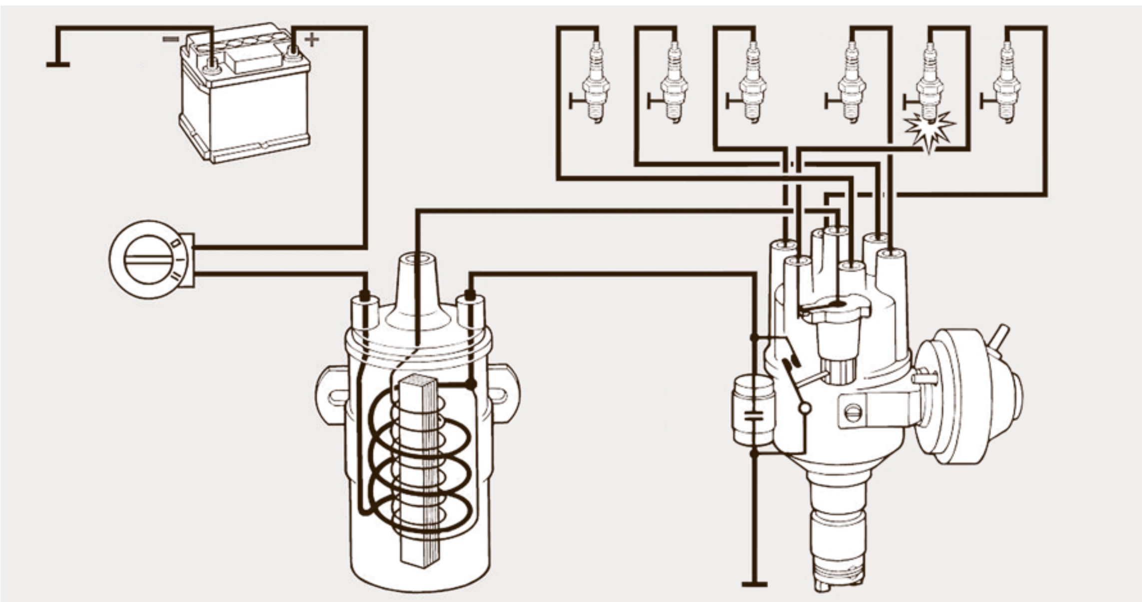



**Fig. 10.89** Conventional coil ignition

A mechanical advance and vacuum unit define the proper ignition point as a function of engine speed and load.

*Electronic Ignition.* This system requires no centrifugal or vacuum-based timing control. Sensors monitor engine speed and load; these signals are sent to the ignition control unit, which determines the optimal ignition timing.

Engine speed is sensed by an inductive pulse sensor; engine load is sensed by a pressure (vacuum) sensor connected to the intake manifold. The ignition control unit uses these signals in a timing program map to determine optimal ignition timing for each speed/load point. Additional sensors may be used to monitor engine temperature, intake air temperature, throttle-plate position, and other operating parameters which are then taken into consideration when determining ignition timing.

This system may utilize a distributor to contain the engine speed/position sensor and to distribute the high voltage (Fig. 10.90a, or may instead be distributor-less (Fig. 10.90b. The latter system has a separate ignition coil for each cylinder or pair of cylinders. These coils may be mounted on the engine or directly attached to the spark plugs.

The ignition control system may be combined with the fuel injection control system to allow for coordinated total engine management (see Sect. 10.4.4).

### High-Voltage Generation

The high voltage needed to bridge the spark-plug gap is typically generated by a coil. The coil consists of two copper windings (primary and secondary), an iron core, and a plastic casing. Energy is transferred from the primary winding to the secondary winding by means of magnetic induction. The current and voltage amplification from primary to secondary is the ratio of the number of coil windings.

Charging and discharging the coil to generate the high voltage needed for the spark plugs is essentially the same for all types of ignitions. The trigger closes and opens once for each cylinder's combustion event. When the trigger closes, current flows through the coil's primary winding and to ground. This produces a flux field in which ignition energy is stored. The time available for charging is determined by how long the trigger is closed (referred to as the dwell angle). The current is interrupted when the trigger opens.

The flux field in the primary winding produces high-tension voltage in the secondary winding. For a distributor-type system, this voltage is conducted to a center contact in the distributor cap. A rotor within the distributor rotates with engine rotation and distributes

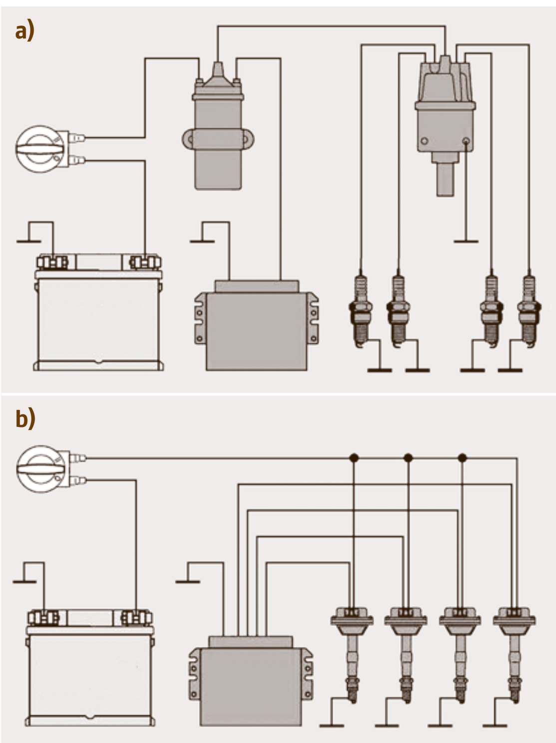



**Fig. 10.90a,b** Electronic ignition

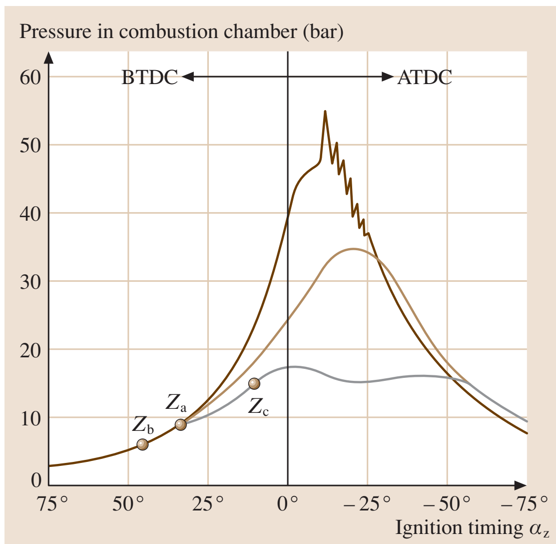



**Fig. 10.91** Effect of ignition timing on combustion pressure

the high voltage to the spark plugs via the distributor cap and ignition wires. For a distributor-less system, the voltage is fed directly to the spark plug.

### Ignition Timing

As engine speed increases, ignition timing must be advanced relative to piston top dead center to allow the air–fuel mixture more time to burn. As load increases, ignition timing must be retarded to prevent detonation (knock). The effect of ignition timing on combustion pressure is illustrated in Fig. 10.91.

### Spark Plug Design

All ignition system designs include a spark plug, which consists of a:

- Terminal post which leads the current from the ignition wires to the center electrode
- Insulator which is made from a ceramic material and insulates the center electrode and terminal post from the shell
- Shell which houses the insulator and allows mounting to the cylinder head
- Gasket and seat which seals the combustion pressure
- Electrodes that form a gap that the high voltage must bridge to pass to ground, thus causing a spark

Electrodes may be made from a compound of corrosion-resistant nickel and copper, a composite with silver as a base, or platinum. The electrode gap determines the length of spark; the voltage required to jump the gap increases as gap width increases.

Figure 10.92 shows a spark plug cross section of two different electrode designs (front electrode and side electrode).

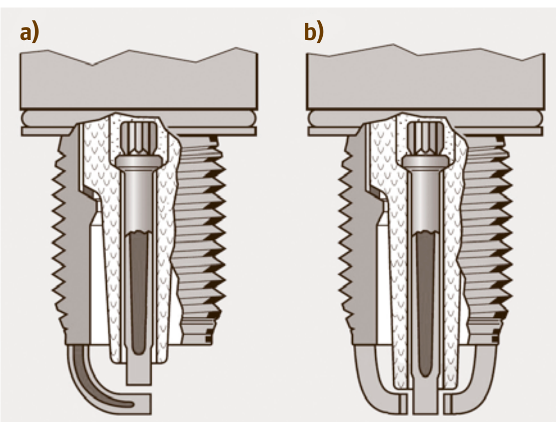



**Fig. 10.92a,b** Electrode designs

The heat range defines the operating temperature of the spark plug's insulator nose and should be chosen to maintain 500–900 °C. If the temperature is below 500 °C combustion residue will form on the insulator nose. If the temperature is above 900 °C the hot combustion residue gases promote electrode oxidation. Above 1100 °C auto-ignition may result. Figure 10.93 shows the proper operating range.

## 10.4.6 Mixture Formation and Combustion Processes

All fuel–air mixtures have limits for the mixture ratios that can be used in engines. When there is too little fuel, the mixture is said to be *lean* and flame propagation is slow and misfire is likely. When the fuel concentration is too high, the mixture is said to be *rich*, and the combustion produces products of incomplete combustion such as carbon monoxide.

### Spark Ignition Engines

In spark-ignited engines, a homogeneous charge of fuel and air is introduced into the cylinder and a flame is initiated with a spark near the end of the compression

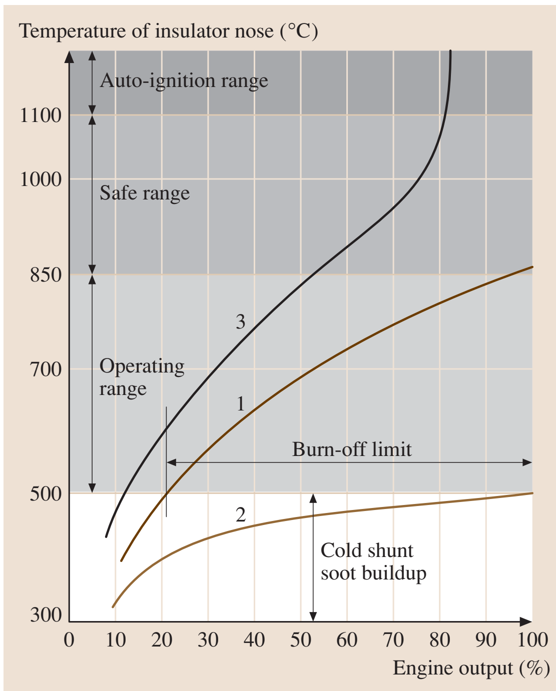



**Fig. 10.93** Operating temperature of the spark plug's insulator nose

process. The flame propagates outward from the spark source. Turbulence is required for the flame to achieve the speed necessary to complete the combustion process before the exhaust valve opens. In the early part of the combustion process, immediately after ignition, the flame growth is slow. The combustion zone, called the flame kernel, is small and very little energy has been released. The progress of combustion at this point is sensitive to the balance of energy released by combustion and the heat lost to the cooler surroundings. Leaner mixtures and lower temperatures and pressures can slow the energy release rate and increase the likelihood of misfire. Cycle-to-cycle variations in this critical phase of the combustion process can cause large differences in the development of the cylinder pressure.

In some cases the combustion process may not proceed in the normal manner of a deflagration wave propagating away from the spark. In one such case, the gases in front of the flame front, which are compressed as the cylinder pressure rises from combustion of the gases behind the flame front, may spontaneously ignite. This *auto-ignition*, also called *knock*, can be extremely violent and significantly raises engine noise levels. If a large percentage of fuel is involved in the event, it can cause mechanical damage to the engine. Some fuels are more prone to auto-ignition than others. The resistance of the fuel to auto-ignition is characterized by the fuel's octane number. Fuels with high octane numbers can be operated with less chance of knock.

Basic thermodynamics indicates that engines with high compression ratio should have better fuel economy and performance. However, the higher temperatures and pressures that result from the higher compression ratio increase the tendency for knock. High octane fuels are needed for engines with elevated compression ratios. Because knocking combustion is so rapid, it tends to approximate the thermodynamic ideal of constant volume combustion. In fact, many engines demonstrate their highest efficiency when operated with light knock. However, as more fuel is consumed by the auto-ignition, the pressure oscillations in the cylinder tend to disrupt the thermal boundary layers in the cylinder and increase the heat loss from the combustion gases.

Another type of abnormal combustion is known as pre-ignition or surface ignition. In this case, the fuel–air mixture may be ignited by a hot surface in the cylinder. This might be a valve surface, a carbon deposit, or even a piece of head gasket that protrudes into the cylinder. Because the combustion may be earlier than the spark, the flame-caused pressure rise may occur during the compression process causing the cylinder pressure to become very high. It has been proposed that knock and pre-ignition can be coupled. Knocking can increase heat transfer rates and raise surface temperatures so that pre-ignition is more likely. The resulting pre-ignition raises the gas temperatures in the cylinder and makes knock more likely.

When the fuel–air mixture is too lean, the flame kernel grows slowly and there is an increased probability of misfire in the engine. The mixture ratio where the misfire becomes unacceptable is the lean operating limit of the engine. There is a corresponding rich operating limit where there is insufficient oxygen to sustain combustion. The chemically correct mixture is known as the *stoichiometric* mixture. The air–fuel ratio for maximum engine power falls on the rich side of the stoichiometric mixture and the best fuel economy tends to be on the lean side.

In order for a spark-ignited engine to operate continuously, the air–fuel mixture must be between the two flammability limits. However, exhaust emission concerns dictate that the mixture be held close to stoi-

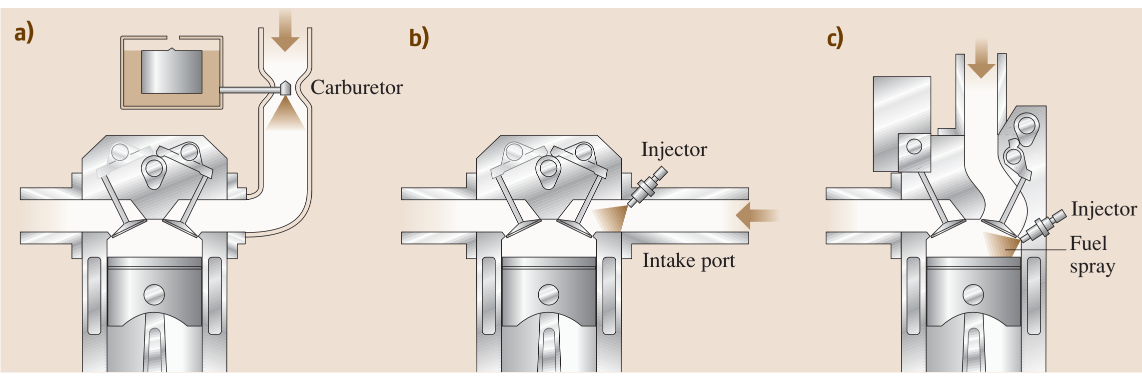



**Fig. 10.94a–c** Fuel–air mixture formation in spark ignited engines

chiometric. Except for cold starts and brief periods of high power demand, modern vehicles equipped with three-way catalysts and closed loop control of air–fuel ratio operate in a narrow band around stoichiometric.

Figure 10.94 shows three different approaches to fuel–air mixture formation in spark-ignited engines. The traditional approach of introducing the fuel with a carburetor into the intake air stream is only used for small engines and is obsolete for larger engines that are subject to emissions controls. Most engines today use port fuel injection, as shown in Fig. 10.94b. This approach provides very uniform air–fuel mixture between cylinders and excellent atomization of the fuel at all speeds. Figure 10.94c shows direct injection of the fuel into the cylinder. Although not yet widely used, this approach allows some degree of charge stratification in the cylinder and fuel economy that approaches the diesel engine.

#### Compression Ignition Engines

*Compression ignition* engines, also known as *diesel* engines, bring only air into the cylinder through the intake valve. The engines rely on compression of the air to produce sufficient temperature that the fuel auto-ignites soon after it is injected near the end of the compression process. In contrast to the homogeneous charge spark-ignited engine, the air–fuel mixture in the diesel engine is always heterogeneous. Since there is a distribution of fuel–air ratios ranging from very lean to very rich, there is always some location in the cylinder where conditions are optimum for auto-ignition and the cycle-to-cycle variability for diesel engines is very small.

Diesel engines run without throttles so they have the advantage of low pumping losses. Load is controlled by varying the amount of fuel that is injected into the cylinder. At light load and idle, the air–fuel ratio may be 70:1 or higher. At full load, the air–fuel ratio may be as low as 20:1. The stoichiometric ratio of 14.5–14.8 is generally not achieved because of high smoke levels.

Diesel engines can be categorized into direct injection (DI) and indirect injection (IDI), although indirect injection designs are now mostly obsolete. The IDI engines utilized a small chamber apart from the main chamber, called a prechamber or turbulence chamber, that was connected to the main chamber by a narrow passageway. Fuel was injected either into the separate chamber, or into the passageway, and the rapid air motion between the two chambers caused by the piston motion, provided excellent fuel–air mixing. This rapid mixing allowed high-speed operation of the engines but heat transfer and throttling losses exacted a severe fuel-economy penalty on IDI engines. Advances in fuel injection technology have allowed DI engines to operate at equivalent speeds with much better fuel economy. A DI engine with the fuel sprayed directly into a chamber in the cylinder formed by a toroidal recess in the piston is the most common configuration in modern diesel engines.

#### Alternative Combustion Systems

A major drawback of both spark ignited and compression ignited engines is their high $NO_x$ emissions. Both engines require after-treatment to reduce $NO_x$ to acceptable levels. Combustion systems that are homogeneous charge, like spark-ignited engines, but utilize auto-ignition, like a compression-ignited engine, have been developed. These engines utilize an air–fuel mixture that would ordinarily be quite lean, but by controlling the temperature, it can be made to auto-ignite in a gradual and controlled manner towards the end of the compression stroke. Temperature and air–fuel ratio can be optimized to reduce $NO_x$ emissions to very low levels.

### 10.4.7 Fuels

This section will cover the basics of gasoline, diesel, and alternate fuels. It will start with the original of petrochemical fuels and the refining process. The basic composition and key characteristics of gasoline and diesel fuels will then be explained. Lastly an overview of fuel substitutes and alternate fuels will be given, whereby this cannot cover in detail the plethora of newly emerging fuels.

#### Petroleum Refining and Basic Organic Chemistry [10.25]

Most conventional fuels are made from petroleum crude oils, consisting primarily of paraffinic, naphthenic, and aromatic hydrocarbons. Raw crude oils have a wide range of densities ranging from as thin as water to as thick as tar. Crude oil is converted into usable products by means of refining; the most important products are gasoline, jet fuel, and diesel fuel. Other valuable products are heating oils, liquefied petroleum gas, lubricating oils and asphalt.

To convert crude oil the feedstock is typically distilled. Since the different components of crude oil (e.g., gasoline, diesel) have different boiling points, the lighter components (those with relatively low boiling points, e.g., propane and butane) rise to the top of the distillation column where they are drawn off. The next-heavier components (e.g., gasoline) are drawn off lower

on the column, then the subsequently heavier components (kerosene and then diesel) are drawn off towards the bottom.

The fuels must then be upgraded, usually by hydroprocessing (which uses hydrogen with a catalyst) to remove undesired components.

Fuels with higher boiling points are then cracked (broken down) into lower boiling points using very high temperatures and catalysts.

### Gasoline

*Basic Composition.* Gasoline fuels for spark-ignition engines are hydrocarbon compounds, which sometimes contain oxygenous components to enhance performance.

*Key Characteristics.*

- Grade: usually stated as regular or premium; an indication of anti-knock property.
- Octane number: resistance to knock (pre-ignition).
- Density: weight per unit volume; energy content increases as density increases.
- Volatility: how easily the fuel vaporizes. The fuel must vaporize quickly for good cold starting but not so quickly as to cause vapor-lock. Volatility is characterized by the fuel's vapor pressure and/or evaporation points dependant upon temperature.
- Sulfur content: must be kept low to allow proper operation of the catalytic converter or other after-treatment device.
- Additives: may be used to enhance one or more of the properties stated above, or to protect against aging, contamination or corrosion.

### Diesel

*Basic Composition.* Diesel fuels for compression-ignition engines are usually distilled from crude oil. They consist of a large number of different hydrocarbon compounds including $n$-paraffins, olefins, naphthenes and aromatic compounds. Diesel fuel ignites at $\approx 350\,°C$, much lower than gasoline, which ignites at $\approx 500\,°C$.

*Key Characteristics.*

- Grade: the standard to which the fuel must conform.
- Density: weight per unit volume; energy content increases as density increases.
- Viscosity: resistance to flow; low viscosity leads to leakage losses, while high viscosity may impair injection pump function.
- Cetane number: ease with which fuel ignites; combustibility increases as cetane number increases.
- Cold filter plugging point: the temperature at which the fuel clogs the filter.
- Flash point: the storage temperature at which flammable vapors are produced.
- Water content: amount of water in fuel; water causes corrosion and poor lubrication, leading to wear and seizures.
- Contaminants: foreign particles in fuel; the particles cause erosive and abrasive wear.
- Lubricity: measure of the fuel's lubrication properties; low lubricity causes wear and seizures.
- Sulfur content: amount of sulfur in fuel; sulfur does not harm the fuel injection system but will harm most after-treatment devices. The removal of sulfur by hydrogenation also removes the ionic fuel components that aid lubrication, reducing the lubricity properties of the fuel. Additives are thus needed to restore lubricity to a sufficient level.
- Oxidation stability: resistance to forming acids.
- Additives: may be used to enhance one or more of the properties stated above.

### Alternate Fuels [10.26]

In addition to the standard fuels there are several alternatives to gasoline and diesel. These are often pursued in order to reduce emissions or to reduce consumption of nonrenewable fuels as most alternate fuels are from a renewable source.

It should be noted that alternate fuels are not always compatible with the fuel system (some require extensive modifications), and that they may increase one emission component while reducing another.

Alternate fuels can be divided basically into two categories: gasoline substitutes/additives and diesel substitutes/additives.

*Gasoline Substitutes/Additives.*

- Coal hydrogenation: coal and coke
- Liquefied petroleum gas (LPG): hydrocarbon mixtures (mostly propane) that are liquid at ambient temperatures under relatively low pressures
- Liquefied natural gas (LNG): methane that is cooled to $< -160\,°C$ and condensed to a liquid by compression
- Compressed natural gas (CNG): natural gas (mostly methane) compressed to high pressure
- Hydrogen: produced by electrolysis of water or from natural gas/coal

- Alcohol: methanol (usually made from natural gas but can also be made from biomass resources such as wood) and ethanol (made from grain or biomass). Either can be used alone or mixed with gasoline

*Diesel Substitutes.*

- Alcohol: methanol and ethanol as described above; usually mixed with diesel.
- Fatty acid methyl ester (FAME): often known as biodiesel, made from many sources the most popular of which are Rapeseed, soybean, sunflower and used cooking oils. Usually mixed with diesel in 5–50% concentrations, these bring lower PM emissions and improve lubricity but are critical regarding density, $NO_x$, water absorption, and stability.
- Diesel–water emulsions: reduce soot and $NO_x$ but also lower power output
- dimethylether (DME): a gas-phase fuel, requiring extensive modification of the fuel injection equipment
- Oil sand/tar sand gasification
- Synthetic fuels: methane gas-to-liquid (GTL, tradename *Synfuel*), biomass-to-liquid (BTL, tradename *Sunfuel*), and coal-to-liquid (CTL).

### 10.4.8 Emissions

Emissions from engines are generally thought of as the harmful chemicals that are present in small amounts in the exhaust gas. However, vaporized fuel that is released during refueling or from leaks onboard the vehicle can also be a source of air pollution. Carbon dioxide, a major component of engine exhaust and an inevitable consequence of hydrocarbon combustion, was traditionally considered to be benign and not included as a hazardous air pollutant. Now that the role of carbon dioxide from fossil resources in global climate change has been identified, much more attention is being paid to this gas.

Emission regulations can be placed into two categories: those that limit emissions from specific engines or vehicles and those that limit the levels of emissions in specific spatial locations, without regard to their origin. The first category of regulations will be our primary focus as these are the regulations that engine and vehicle manufacturers are required to meet in order to sell their products. The second category of regulations tend to be the responsibility of cities or states, and while they may impose restrictions on the types of vehicles that can be sold within their jurisdiction, the sources of pollution are not limited to engines and vehicles.

#### Ambient Air Quality

Ozone has been the subject of what appear to be contradictory statements in the popular media. Ozone is described as essential to protect against skin cancer and is being depleted by chlorofluorocarbons (CFCs). At the same time, city dwellers are subject to warnings about

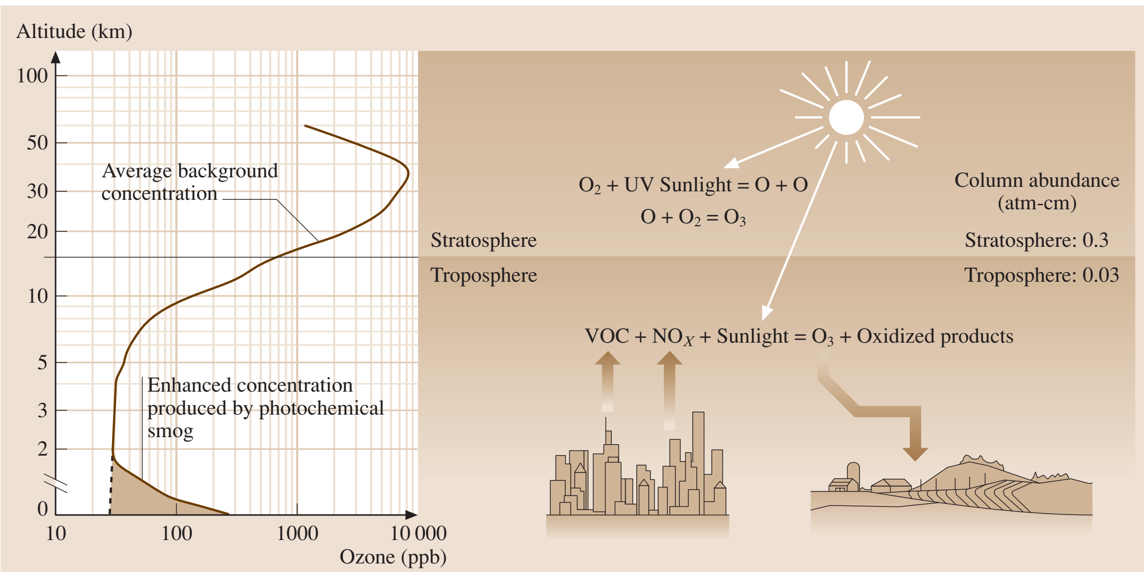



**Fig. 10.95** Atmospheric ozone (after [10.27])

high ozone levels. The resolution of this apparent contradiction lies with understanding the role of altitude as explained in Fig. 10.95. In the stratosphere, above 15 000 feet, the concentration of ozone can be very high, approaching 10 000 ppb (parts per billion). This ozone is formed by reactions involving sunlight and oxygen. It filters the ultraviolet solar radiation that causes skin cancer. Some researchers have argued that this ozone is destroyed by reactions with chlorine atoms originating from CFCs.

At lower altitudes the natural concentration of ozone is up to 3 orders of magnitude lower than in the stratosphere. This is fortunate since high levels of ozone interfere with plant growth and are a strong irritant. High levels of ozone can be produced at lower elevations by reaction of volatile organic compounds (VOCs), carbon monoxide, oxides of nitrogen ($NO_x$), and sunlight. This complex set of chemical reactions produces a large number of different chemical compounds, many of which are harmful and irritating to people. Ozone, although only one chemical compound, is widely used as a measure of the overall concentration of the complex chemical mixture, sometimes known as *smog*. The EPA recently reduced the maximum allowable ozone concentration from 120 ppb averaged over a 1 h period to 80 ppb averaged over an 8 h period. Normal background levels of ozone are typically 20–40 ppb and can exceed 200 ppb during severe smog episodes [10.24].

Ozone is a secondary pollutant. It is not found in significant amounts in the exhaust of engines. However, compounds that are found in engine exhaust contribute to the formation of ozone, such as, VOCs, carbon monoxide, and $NO_x$. Another major source of VOCs is evaporative emissions. Evaporative emissions originate from losses of fuel during refueling as well as when the vehicle undergoes diurnal heating and cooling. These emissions are strongly tied to the Reid vapor pressure of the fuel, which is why it is now closely controlled in areas where air pollution is a problem. Evaporative emissions are only a problem with volatile fuels such as gasoline and its blends with alcohol. Diesel fuel's vapor pressure is so low that it does not contribute to evaporative emissions.

### Regulated Pollutants

The Environmental Protection Agency (EPA) regulates the tailpipe emissions of both spark-ignition and compression-ignition engines in the United States. Regulated pollutants from spark-ignited engines include carbon monoxide, oxides of nitrogen, and unburned hydrocarbons. Compression ignited, or diesel, engines must meet requirements for particulates as well as these gaseous species.

Carbon monoxide is primarily determined by the engine's air–fuel ratio. When the engine is operated fuel-rich there is insufficient oxygen to convert all of the carbon to carbon dioxide, so a portion is converted to carbon monoxide. Carbon monoxide is actually an intermediate product in the oxidation of hydrocarbons and is always present in significant amounts during the combustion process. Measured levels in the exhaust are usually higher than would be expected because the oxidation of the carbon monoxide tends to be a slow process that is limited by the rate of the reaction of CO with the OH radical, as shown in the following equation

$$CO + OH \rightarrow CO_2 + H\,. \tag{10.82}$$

The concentration of the OH radical decreases rapidly as the in-cylinder temperature drops during expansion, leaving the CO frozen at an elevated level. An additional mechanism that affects homogeneous charge engines is the partial oxidation of trapped fuel that emerges from crevices or oil films during the expansion process when the temperature is too low to oxidize the fuel completely before the exhaust valve opens. In carbureted engines, rich-burning cylinders resulting from the nonuniform distribution of fuel between cylinders is an important source of carbon monoxide. Carbon monoxide emissions from diesel engines are generally well below regulation limits because diesels always operate with excess air.

Oxides of nitrogen ($NO_x$) consist primarily of nitric oxide (NO) and nitrogen dioxide ($NO_2$). Nitric oxide originates through three potential mechanisms that are usually categorized as fuel $NO_x$, prompt $NO_x$, and thermal $NO_x$. Fuel nitrogen can contribute to $NO_x$ formation but is usually not important for engines because gasoline and diesel fuel contain small amounts of nitrogen. Prompt $NO_x$ is formed by reactions between nitrogen and hydrocarbons during the combustion process. This mechanism also does not seem to be an important source of $NO_x$ for engines. The primary source of engine $NO_x$ emissions is thermal $NO_x$. This mechanism involves the following three reactions

$$O + N_2 \rightarrow NO + N\,, \tag{10.83}$$

$$N + O_2 \rightarrow NO + O\,, \tag{10.84}$$

$$N + OH \rightarrow NO + H\,. \tag{10.85}$$

Since these reactions require significant concentrations of the radicals O, N, and OH, they only occur at

high temperatures. The reactions also require significant time to equilibrate so most of the NO formation occurs in the post-flame gases. Virtually all $NO_x$ control strategies, such as timing retard and exhaust gas recirculation (EGR) focus on reducing the temperature of the post flame gases.

Unburned hydrocarbon emissions from spark-ignited engines generally originate from fuel that is trapped in crevices, oil films, or deposits and is thus protected from combustion during the main combustion event. This sequestered fuel is released when the pressure drops during the expansion process but the temperature may be too low for complete combustion. Some of the fuel may burn to carbon monoxide but a significant portion will remain unburned or only partially burned and this will be released in the engine exhaust. Some of the products of partial combustion, such as olefins and aldehydes, are highly reactive and are strong contributors to photochemical smog reactions. Occasional misfiring cycles can also be a significant source of unburned hydrocarbon (UHC) from spark-ignited engines. UHC from diesel engines generally originate from fuel that has been overmixed with air so that the mixture is too lean to burn under the conditions in the cylinder. These conditions are most likely to be encountered at idle and light loads.

Regulated emissions from compression-ignited, or diesel, engines include the CO, $NO_x$, and UHC described for spark ignition (SI) engines, but also include particulates. Particulates from diesel engines are operationally defined as whatever collects on a filter when the exhaust is cooled to 52 °C after the filter has had a chance to equilibrate in a temperature and humidity controlled environment. The primary constituent is carbonaceous matter, usually referred to as soot, that originates from high temperature pyrolysis reactions in the fuel-rich regions of the cylinder. The carbonaceous particles provide sites for the condensation and adsorption of high molecular weight hydrocarbons as the combustion products cool and this portion of the particulate is often referred to as the soluble organic fraction (SOF) or the volatile organic fraction (VOF). These high molecular weight hydrocarbons may originate from the fuel but are more frequently associated with the lubricating oil. Particulate may also contain sulfates resulting from the reaction of fuel-based sulfur to sulfur trioxide and then to a variety of sulfate compounds which may be observed as small droplets of sulfuric acid. Finally, the particulate may include inorganic compounds resulting from engine wear and lubricant additives. Many of the compounds identified in the SOF are known carcinogens and the small size of the particulates (0.01–0.1 μm) increases the potential for their inhalation and retention in the lungs. For these reasons, the regulatory levels for particulate emissions have been progressively lowered so that after 2007 exhaust filtration technology has been required for most on-highway engines.

### Measurement Instruments

A variety of instrumentation technologies have been developed to quantify the levels of pollutants in engine exhaust gases. Some techniques such as Fourier transform infrared spectral analysis have broad applicability and are widely used for engine development. For emissions certification, specialized instruments are still used that have been developed to measure specific species, and often over limited ranges. Oxides of nitrogen are measured with devices that take advantage of the chemiluminescent reaction that occurs when NO reacts with ozone to form $NO_2$ and oxygen. The photon of light that is emitted by this chemical reaction can be measured and directly related to the concentration of NO. Total $NO_x$ can be measured by passing the exhaust gas through a catalyst that converts the $NO_2$ into NO before the gas is exposed to ozone.

Carbon monoxide and carbon dioxide are most commonly measured with nondispersive infrared (NDIR) absorption instruments that measure the amount of light of a specific wavelength that is absorbed by the exhaust gas. The wavelengths and path lengths for the light are chosen to provide the best sensitivity for the gas of interest. This technique requires that water vapor, a broad-band absorber of infrared radiation, be removed before the measurement can be performed. The water is usually removed by cooling the gas to condense the water or passing the gas stream through a chemical desiccant.

Unburned hydrocarbons are measured using a flame ionization detector. These devices contain a small hydrogen flame located between two electrically charged plates. A small amount of the exhaust stream is fed into the hydrogen flame and the hydrocarbon-based carbon atoms produce flame ionization that can be measured as an electric current between the charged plates. The particle filters and all connecting lines must be heated to prevent condensation of the hydrocarbon vapors before they enter the flame. The heated flame ionization detector (HFID) measures the number of carbon atoms associated with hydrocarbons in the exhaust and thus requires an assumption to be made regarding the chemical composition of the hydrocarbons. Measurements

performed on gasoline-fueled engines often assume the hydrocarbon has the same structure as hexane. Assuming the unburned hydrocarbons have the same chemical structure as the fuel is also a common assumption for both gasoline and diesel engines.

Particulates are measured by filtering a portion of the exhaust gas and then weighing the increase in mass of the filter. The temperature of the filter is carefully controlled because if it is too low an excessive amount of the unburned hydrocarbon vapors may condense on the filter. The techniques used to capture a representative sample of the exhaust gas and allow the determination of the amount of particulate during a transient test cycle are described in the following section.

### Test Cycles

Emissions from passenger cars and other light-duty vehicles are measured while the vehicle is operated on a chassis dynamometer. The chassis dynamometer connects the drive wheels of the vehicle to a load absorber through a set of rollers. The device allows the vehicle to be operated in a controlled laboratory but with an accurate simulation of actual in-use driving conditions. Flywheels and dissipative absorbers are used to simulate vehicle inertia and air resistance so the vehicle can be operated over transient driving cycles. Test cycles that involve following a vehicle speed versus time curve that models different driving conditions are used for emissions certification. Trained drivers can follow these curves very exactly although most installations now use computerized throttle controllers.

The wide variety of transmission, drive line, and engine combinations used in heavy-duty applications precludes the emissions certification of vehicles. Heavy duty emissions testing focuses on testing the engine itself while it is outside the vehicle connected to a computer-controlled load absorber. The engine is operated over a 20 min cycle where both the engine speed and torque are specified on a second-by-second basis. The 20 min test cycle consists of four 5-min segments that model different types of city and highway driving.

Exhaust emissions for both chassis dynamometer and engine dynamometer test systems involve injecting some or all of the exhaust gas stream into a dilution tunnel to lower the temperature of the exhaust gas and to simulate the particulate agglomeration processes that occur when the exhaust enters the atmosphere. These systems are equipped with flow control systems so that a constant volumetric flow rate is maintained for the sum of the engine exhaust and the dilution air. A sample of the diluted exhaust gas is filtered and the weight of the particulate is measured. In the case of chassis dynamometers, the pollutant is expressed as g/mile and for the engine dynamometer it is expressed as g/horsepower-hour or g/kW h. The denominator of the engine dynamometer term is the total amount of work performed by the engine during the test cycle.

Gaseous emissions are also sampled from the dilution tunnel. They may be collected in special chemically inert bags to obtain an integrated total for the cycle, or measured second by second to investigate the effect of different parts of the test cycle on a pollutant of interest.

Both chassis and engine dynamometer testing are conducted under carefully controlled laboratory conditions. There have been some claims that this testing is not representative of emissions from in-use vehicles and attempts have been made to characterize in-use emissions using chassis dynamometers to test vehicles chosen at random from traffic flows. This experience has shown that significant numbers of vehicles have improperly operating emission control systems and actual emission levels are higher than would be expected from emission certification data. Measurements have been attempted using light absorption techniques from passing traffic but results have been mixed. Variability in measurement points, vehicle types, weather effects, etc. mean that extremely large numbers of measurements are required.

### SI Engine Emissions Characteristics

The primary pollutants of regulatory concern for spark-ignited engines are carbon monoxide, oxides of nitrogen, and unburned hydrocarbons. The levels of these pollutants in the engine exhaust depend strongly on the engine's operating conditions such as spark timing, load, speed, and air–fuel ratio. While the engine speed and load are controlled by the operator (for a manual transmission), the timing and air–fuel ratio are set by the engine's electronic control module (ECM) to keep the levels of the exhaust pollutants within the range allowed by emissions regulations while ensuring adequate vehicle performance. Under cold starting and high-power-demand conditions the air–fuel ratio is calibrated to be fuel-rich. However, under all other operating conditions, the ECM maintains the air–fuel ratio close to the chemically correct, or stoichiometric, ratio.

The impact of timing, speed and most other operating parameters on carbon monoxide is secondary to the air–fuel ratio. There may be some effect on CO as these parameters are varied but this is most likely due to changes in air–fuel ratio or in causing the engine to operate at non-optimum conditions for oxidation of the CO.

As described above, oxides of nitrogen from spark-ignited engines are almost entirely thermally based and the two primary parameters that influence combustion temperature are spark timing and air–fuel ratio. Over the normal range of variation of spark timing used in modern engines, earlier timing (advanced) always increases emissions of oxides of nitrogen ($NO_x$) and later timing (retarded) always decreases $NO_x$. Since the spark timing that provides best fuel economy is usually well advanced from the timing needed to keep $NO_x$ at a tolerable level, engine designers are confronted by a trade-off between the desire to provide good fuel economy while still keeping $NO_x$ emissions at a level that can be controlled by the catalytic converter.

While spark-ignited engine UHC emissions are primarily dependent on design factors such as piston ring position, which controls the size of the crevice volume, they can be affected by speed, timing, load, and fuel–air ratio as these will affect the amount of fuel that is sequestered in crevices and the tendency to misfire. As mentioned earlier, evaporative emissions are another source of UHC from spark-ignited engines. While these can be controlled during engine operation by venting the fuel tank through a carbon canister to absorb fuel vapors, vehicle refueling still usually involves a release of vapors as capture systems are not required in most states. Evaporative emissions are best controlled by limiting the fuel's vapor pressure during warm weather.

In some parts of the United States, oxygenates are required to be added to gasoline to lower emissions. Ethanol is currently the most widely used fuel oxygenate. Methyl *t*-butyl ether (MTBE) is being rapidly phased out due to concerns about groundwater contamination resulting from fuel spills. Because of its lower oxygen requirement for combustion, ethanol decreases emissions of CO and UHC during cold starting and periods of high load. This is partially offset by higher evaporative emissions resulting from the higher vapor pressure when ethanol is blended with standard grades of unleaded gasoline.

*Closed-Loop and Open-Loop Control.* As described above, the pollutant species that need to be controlled from spark-ignited engines are carbon monoxide, unburned hydrocarbons, and oxides of nitrogen. It should be noted that the first two of these compounds need to have their oxidation process completed, while the third compound needs to be *un-oxidized* or reduced. This conflict in objectives is what makes the simultaneous elimination of the three compounds so difficult. Completion of the oxidation process can be accomplished by providing an excess of oxygen and then passing the exhaust gas through an oxidizing catalyst such as platinum. Reducing the oxides of nitrogen requires an oxygen-poor environment and an easily oxidized compound, called a reducing agent, to assist in breaking down the nitric oxide.

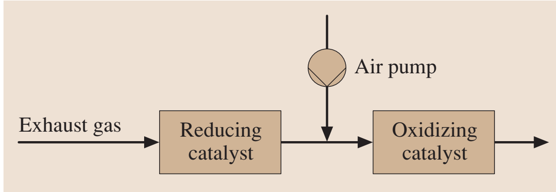



**Fig. 10.96** Two-part catalytic converter

Early technology to achieve simultaneous control of the three pollutants required two operations, as shown in Fig. 10.96.

The engine was operated somewhat rich so that conditions were suitable for the exhaust entering the reducing catalyst, which eliminated most of the nitric oxide. Then an air pump supplied additional air to the exhaust stream to create the oxygen-rich conditions needed by the oxidation catalyst.

Modern catalysts, called three-way catalysts, can combine the oxidation and reduction functions into a single catalyst structure but require that the air–fuel ratio be controlled precisely in a narrow band around the stoichiometric value as shown in Fig. 10.97. Such exact

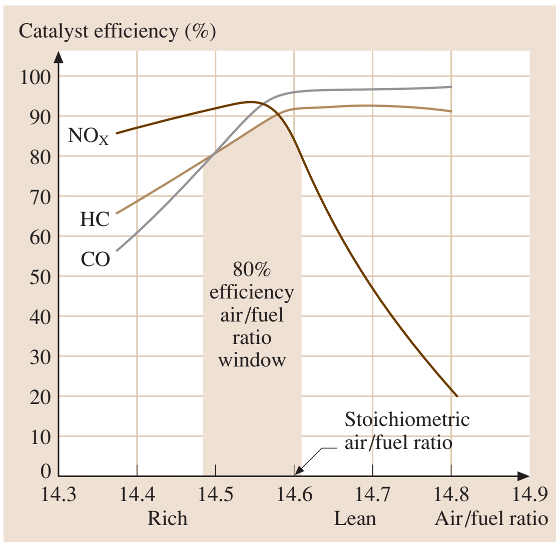



**Fig. 10.97** Effect of air–fuel ratio on a three-way catalyst (after [10.28])

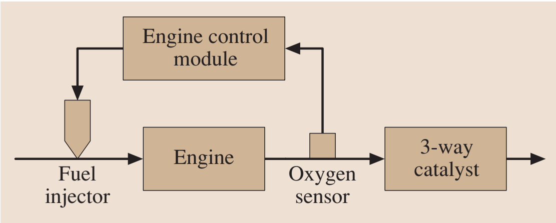



**Fig. 10.98** Closed-loop engine control

control of air–fuel ratio could not be achieved without feedback from an exhaust oxygen sensor and closed-loop electronic control of fuel injection. Figure 10.98 shows the typical configuration of an oxygen sensor that measures the fuel–air ratio in the exhaust gas and then the engine's electronic control module adjusts the fuel injected to correspond to the air flow rate. This system maintains the air–fuel ratio entering the three-way catalyst in a narrow range around the stoichiometric ratio.

In reality, the nature of the oxygen sensor and the delay associated with the time required for changes in air–fuel ratio to reach the oxygen sensor results in an oscillation of the air–fuel ratio around the stoichiometric condition. This oscillation between rich and lean conditions enhances the catalyst's ability to provide the oxygen-poor conditions needed to reduce the oxides of nitrogen while using oxygen stored during the periods of lean operation to eliminate carbon monoxide and unburned hydrocarbons. Recent advances in oxygen sensor technology have incorporated heaters to decrease the time needed for the sensor to reach operating temperature and wideband sensing of air–fuel ratios to provide greater flexibility in control strategies over conventional rich-lean dual state sensors.

A complementary technique for $NO_x$ reduction is exhaust gas recirculation (EGR). This technique directs a portion of the engine's exhaust gas back to the intake where it mixes with and dilutes the incoming charge. It reduces the flame temperature and the availability of oxygen without the fuel economy penalty that would accompany the equivalent $NO_x$ reduction from spark timing retard. Exhaust gas recirculation is most effectively used under part-load conditions where it allows the throttle to be more open and thus can actually improve fuel economy by reducing throttling losses.

### Compression Ignition (CI) Engine Emissions Characteristics

With the exception of carbon monoxide, exhaust emissions from compression ignited engines will be strongly affected by engine operating conditions. Carbon monoxide emissions are always low from diesel engines. They tend to be limited by factors such as late burning fuel that has insufficient time or temperature to combust completely. While more fuel is present at high load, the temperatures are lower at light load and there is a greater availability of fuel that has mixed beyond its lean combustion limit and thus can only react slowly.

Unburned hydrocarbon emissions from diesel engines are primarily a light load problem. At heavy loads the in-cylinder temperatures are high enough that the fuel readily burns to complete products. At light loads, fuel on the periphery of the fuel spray mixes with the large excess of air in the chamber and never reaches the temperature or air–fuel ratio needed for rapid combustion. This fuel will be emitted as unburned or partially burned hydrocarbons. Engines with the greatly retarded timing needed for $NO_x$ control may also have difficulty keeping the fuel–air mixture in the piston bowl because the piston may be well down on its expansion stroke while fuel injection is still underway. This allows the fuel–air mixture to enter the crevice above the top compression ring and evade combustion in manner that is similar to spark-ignited engines.

As mentioned earlier, $NO_x$ emissions are primarily dependent on in-cylinder temperatures. Since these temperatures are higher when the engine is operating at full load, the emissions of $NO_x$ will be higher under these conditions. Diesel fuel injection timing is generally retarded to keep $NO_x$ emissions low, in a manner that is similar to spark timing retard in spark-ignited engines. EGR can be used very effectively to reduce $NO_x$ in diesel engines, especially if the EGR is cooled before it is mixed with the intake air. The cooling is usually accomplished using engine coolant so a portion of the exhaust energy is added to the engine's heat rejection load. EGR is usually accompanied by an increase in exhaust particulates.

Electronically controlled engines can vary the injection timing corresponding to speed, load, or other operating variables following detailed maps. This allows $NO_x$ to be controlled with minimum impact on fuel economy and particulate emissions. $NO_x$ is maximized when the fuel and air are mixed rapidly and combustion occurs near TDC. This is the condition that provides best conditions for minimizing soot production and maximizing its subsequent oxidation. These conflicting effects give rise to the phenomenon known as the $NO_x$-particulate tradeoff. Those measures taken to minimize $NO_x$ (timing retard, EGR, lower swirl, etc.) tend to increase particulates and the converse is also true. Higher fuel injection pressures with smaller injec-

tor nozzle holes and lower intake air temperatures tend to move the $NO_x$–particulate tradeoff to more favorable operating points where both pollutants are reduced.

Exhaust gas recirculation can be problematic with diesel engines because the intake manifold pressure is frequently at a higher pressure than the exhaust manifold. Some systems direct the exhaust from upstream of the turbine to upstream of the compressor, which forces the exhaust gas to pass through the compressor. Other approaches use throttling to lower the pressure of the air entering the compressor so that exhaust can be drawn in from an ambient pressure source after the turbine.

Most diesel engines are equipped with turbochargers that allow demands for increased power to be met while still maintaining the air–fuel ratio at values that ensure low emissions. To meet $NO_x$ emission standards, highly-rated engines use heat exchangers known as intercoolers or aftercoolers to reduce the temperature of the compressed air from the turbocharger compressor. Using ambient air as the exchange fluid for the intercoolers is the norm for most applications. Further improvements in engine air supply can be obtained from the use of variable geometry turbocharging. These turbochargers are equipped with variable area turbine nozzles so the exhaust velocity entering the turbine can be optimally matched to the engine's speed and load. This provides greater intake air boost pressures over a wider range of operating conditions.

Diesel engines cannot use the three-way catalyst technology that is used for spark-ignited engines because diesel engines always operate with excess air and it is difficult to reduce $NO_x$ under lean conditions. Recent developments in catalyst technology have produced lean $NO_x$ catalysts but their low efficiency has limited their acceptance. Catalytic systems that absorb the $NO_x$ and then periodically release it as harmless gases when the engine is momentarily operated rich have been more successful and may be used in the near future.

## 10.4.9 Selected Examples of Combustion Engines

### Compression Ignition Engine with Twin Turbo Technology (BMW 535d)

The world's first use of twin turbo technology for car diesel engines sets this engine apart from comparable engines. Along with increasing the specific power to 67 kW/l, most notably the speed range has been expanded to approximately 5000 $\mathrm{min}^{-1}$. With a bore of 84 mm and a stroke of 90 mm, the engine design is based on the 3.0 l inline-six 530d. The engine's rated output is 200 kW and thus increased by 25% over its predecessor. Just like the power, the maximum torque was raised to 560 N m. The engine weight was increased by 14 kg, while specific consumption at maximum output was reduced by 7 g/kW h to 233 g/kW h compared to engines years older.

*Crankcase and Transmission.* The crankcase cast from pearlitic gray cast iron (GG25+) is based on the *deep skirt concept* already proven in preceding models. The side panels of the crankcase (crankcase skirts) are very deep. A special head design of the skirt area achieves more stiffness. The viscous damper first used in the 530d is used again in the 535d too. The damping effect is generated by varying shear forces in a highly viscose fluid in a narrow gap between the housing and swivel flywheel rim.

*Mixture Formation and Combustion.* The proven concept of BMW direct injection engines has also been adopted in this generation of engine. A central perpendicular injection nozzle and two intake and two exhaust ducts per cylinder are located on the cylinder head. One of the two intake ports is a swirl duct and the other a tangential duct. The two exhaust ducts are still combined in the cylinder head.

To reduce raw emissions, combustion has been concentrated in the outer zones of the piston combustion bowl. Moreover, the engine's compression ratio has been reduced from 17 : 1 (530d) to 16.5 : 1 by further optimizing the piston combustion bowl geometry.

*Injection System.* The injection system of the 535d is based on the second generation common rail system already used in the 530d with a maximum injection pressure of 1600 bar. The flow was elevated in the 530d by 20%. This injection system supports up to five injections with minimal injection intervals between injections per combustion cycle. The induction-side high-pressure pump's volume control generates maximum pressure as required. A micro-blind hole injection nozzle with six spray holes is used. Fuel consumption in the New European Driving Cycle (NEDC) test cycle is 8 l/100 km.

*Exhaust System.* The particle filter is an integral part of the exhaust system of the Euro-4 package. A second-generation filter with catalytically coated SiC substrate is used. With a 4.5 l volumetric capacity, the particle filter has a considerably longer service life than the first generation filter. Two exhaust temperature sensors and

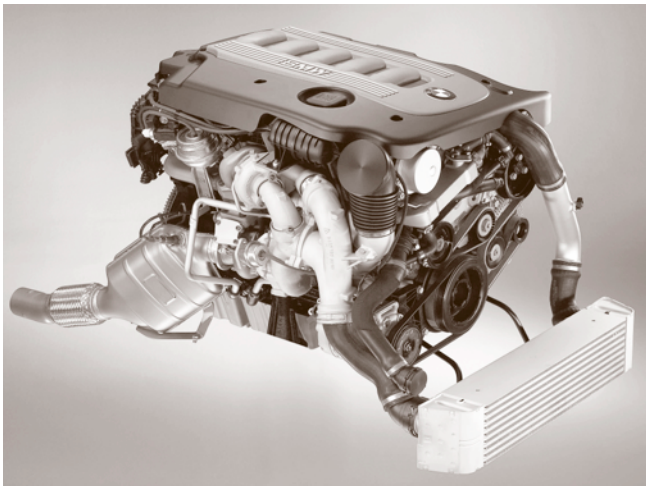


Fig. 10.99 BMW 535d

a pressure sensor are integrated in the exhaust gas line to monitor filter status. A temperature sensor integrated at the inlet of the upstream primary catalytic converter measures the exhaust gas temperature crucial for regenerating the particle filter.

*Operating Principle of Twin Turbo Technology.* Along with different supercharging concepts (multi-stage supercharging, a combination of mechanical and exhaust gas turbo charging) BMW favored the twin exhaust gas turbocharging after several tests.

This system consists of two differently sized turbochargers arranged in the induction and exhaust system branch as illustrated in Fig. 10.100. At lower speed ranges, the compressor bypass and the exhaust butterfly valve are closed as a result of which the entire exhaust mass flow is conducted through the small turbine. In this operating range, only the small turbocharger regulates the supercharging pressure. When the desired supercharging pressure is reached, the exhaust butterfly valve opens. The compressor bypass valve remains closed and part of the exhaust mass flow is conducted to the large turbine. The large compressor functions as a precompressor for the subsequent small compressor, which achieves supercharging pressures at average speeds (maximum supercharging pressure of 2850 mbar at 2500 $\mathrm{min}^{-1}$).

From a certain speed onward, the small compressor can no longer generate additional supercharging pressure and throttles the induction mass flow. Depending on the load and upwards approximately 3000 $\mathrm{min}^{-1}$, the compressor bypass and the exhaust butterfly valve open synchronously so that only the large turbocharger regulates the supercharging pressure supported by the waste gates.

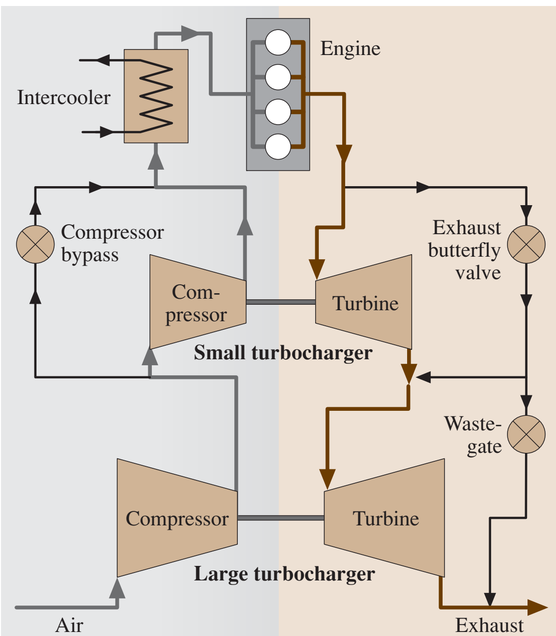



Fig. 10.100 Operating principle of twin turbo technology

### Direct Injection Gasoline Engine with Downsizing-Concept and Dual Supercharging (VW 1.4l TSI)

This engine constitutes a logical contribution to downsizing modern gasoline engines. The direct injection FSI and dual supercharging used for the first time in this form achieve a power of 125 kW from only 1.4 l of displacement. Thus, the compressor, disengageable from the exhaust turbocharger, already reaches the maximum torque of 125 N m at an absolute supercharging pressure of 2.5 bar at low speeds. Along with increasing power, this downsizing concept most notably satisfies the requirement of low consumption of 7.2 l/100 km.

*Basic Engine.* The TSI unit is based on the four cylinder 1.4 l (66 kW) FSI engine used in the Golf V with a bore of 76.5 mm and a stroke of 75.6 mm and a compression ratio of 10 : 1. A basic reason for selecting this engine is the 1.4 l engine's modular design. Many modules could be carried over as a result of which the engineering was limited to a new cylinder crankcase and a water pump with an integrated magnetic clutch for engaging the mechanical supercharger.

The crankcase has an open deck (an open water jacket in the direction of the cylinder head) and deep skirt design (side panels far below the crankcase). Along with the advantage of simpler manufacturing, the open deck variant reduces cylinder barrel deformation when the cylinder head is bolted together. In order to withstand the high mean pressures of 21.7 bar in every operating situation, the material used is GJL (lamellar graphite cast iron), thus achieving a very low weight of 29 kg.

*Transmission.* Above all, great importance was attached to the engine acoustics. As opposed to the 1.4 l 66 kW, a steel crankshaft with 23% more stiffness was used for the TSI. This improves engine sound quality.

Calculation and development tools make it possible to develop a piston for use in a supercharged engine with a specific power of 90 kW/l. This light metal piston's combustion chamber bowl has a pronounce edge to control the flow. In order to provide the piston sufficient operational stability, oil ducts bolted into the main oil gallery inject with approximately 2 bar against the hot outlet side of the piston.

Finally, the piston pin diameter was enlarged because of the considerably higher ignition pressure.

*Injection.* The TSI engine is being used for the first time with a multiple hole, high-pressure injection valve with six fuel outlet bores. The nearly unlimited arrangement of the injection valve's spray makes it possible to form the fuel injection spray. Among other things, this not only optimally homogenizes the mixture but also prevents wetting of the intake valve when there is early injection. This reduces the hydrocarbons (HC) emissions.

The TSI injection pressure raised to 150 bar is generated by an adapted high-pressure pump. Compared to the FSI, its significant features include a longer cam stroke, the use of a roller tappet and the forged aluminum housing, all of which made it pos-

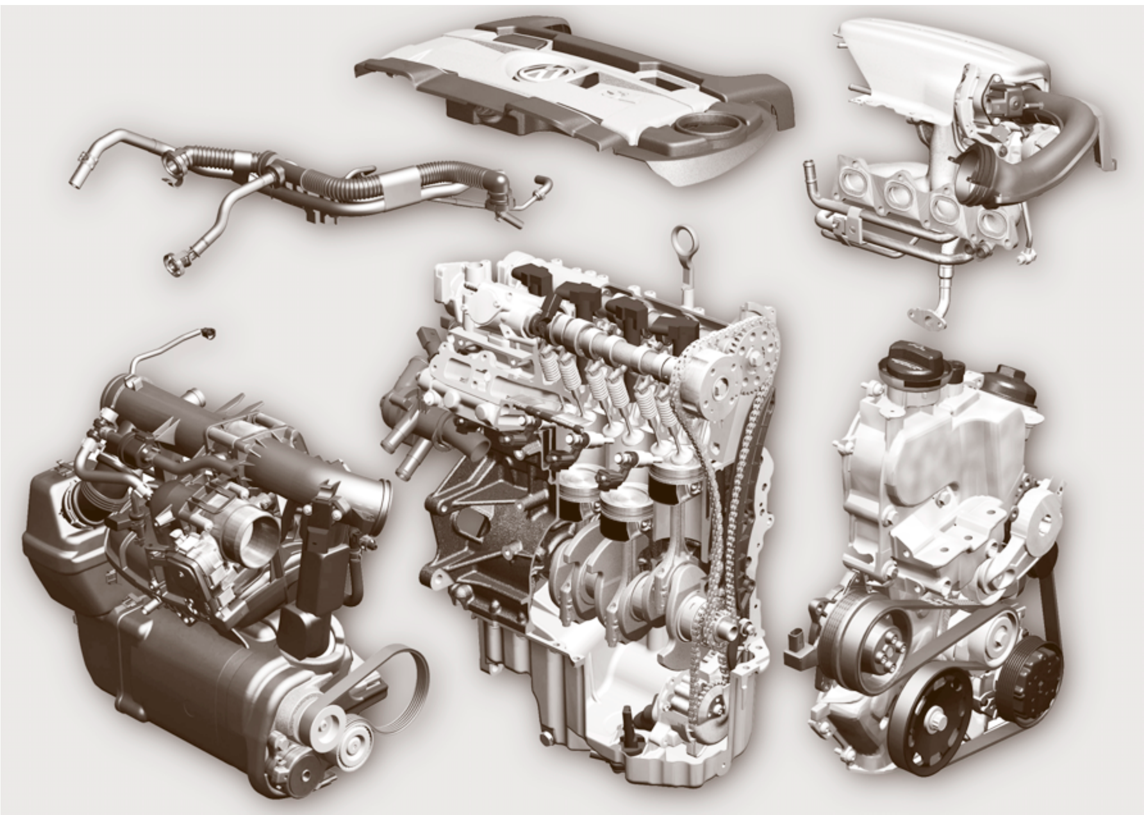


**Fig. 10.101** VW 1.4 l TSI

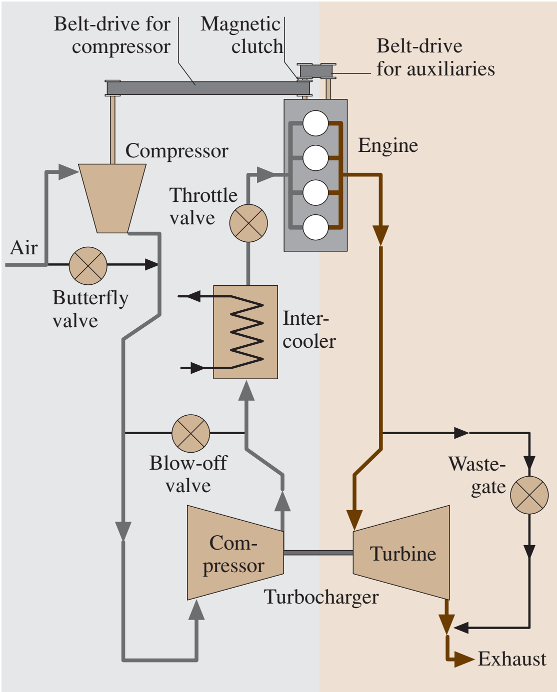



**Fig. 10.102** Principle of twin supercharging

sible to approximately double the pump's mechanical stability.

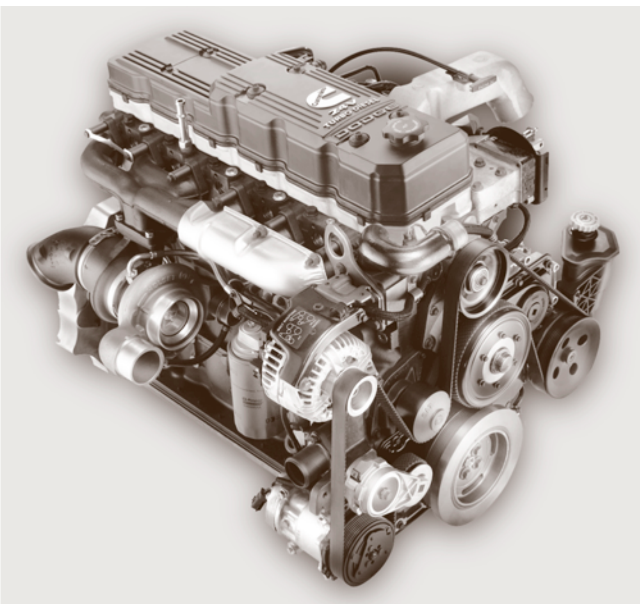


**Fig. 10.103** Cummins 600 turbo diesel

*Supercharging.* Twin supercharging (see Fig. 10.102) basically consists of a Roots supercharger, an exhaust turbocharger and a butterfly valve.

Dependent on the engine map, the magnetic clutch on the water pump connects the compressor to the crankshaft. Inside the compressor is a countershaft transmission, which supplies a high torque above all when starting and in the low speed range. The butterfly valve enables a smooth transition between pure compressor and turbocharger operation.

By using the two charger units, maximum torque can already be produced at 1250–6000 $\mathrm{min}^{-1}$. Since the exhaust gas turbocharger is designed for high efficiency, not enough boost pressure is available in the low speed range. Here, the compressor engages and bypasses the so-called *turbo lag*. At a speed of 3500 $\mathrm{min}^{-1}$, the magnetic clutch disengages the compressor and the butterfly valve opens completely. From this point onward, the exhaust gas turbocharger alone produces the necessary boost pressure.

### Diesel Engine for Heavy-Duty Pickups (Cummins 600 Turbo Diesel)

With 325 HP at 2900 $\mathrm{min}^{-1}$, the Cummins 600 turbo is one of the most powerful engines available for the pickup truck market. Available as the engine in the Dodge Ram heavy-duty pickup, this diesel engine already generates a peak torque of 810 N m (600 ft-lbs) at 1600 $\mathrm{min}^{-1}$. However, weighing 544 kg, the 600 Turbo's average consumption has dropped about 2% below that of the earlier model.

This engine consist of 30% fewer parts than comparable V8 engines in its performance class. Consequently, not only assembly time is cut but fewer repair costs are incurred because of its increased service life and durability.

*Basic Engine and Transmission.* Based on the B series, an agricultural machine engine sold over three million times, the Interact System B (ISB) was developed with an EGR radiator and a supercharging system. Predominantly used in medium-weight commercial vehicles, this engine series is the basic engine for the 600 Turbo diesel. A bore of 102 mm and a stroke of 120 mm produce a displacement of 5.9 l. This inline-six's compression ratio is 17.3 : 1. Push rods control the valves of the bottom-mounted camshaft.

The engine block is manufactured of cast steel and has a correspondingly high stiffness. Along with prolonging service life, this most notably reduces noise emissions.

The transmission is also designed to have a long service life. Not only the crankshaft optimized for weight and stiffness but also the forged fracture-split connection rods contribute to this.

These measures lead to considerably long running times so that an overhaul is only expected after an average of 350 000 miles.

*Induction System and Cylinder Head.* To enable an optimal turbocharge cycle, the Cummins 600 turbo diesel is equipped with four valves per cylinder. The redesigned intercooler as well as the turbocharger's enlarged compressor wheel and housing achieve an optimal air flow during the induction phase.

Reinforced inconel exhaust valves and cobalt stellite exhaust valve seats are used to increase the engine's service life.

*Injection and Combustion.* The 600 turbo diesel engine employs a Bosch high-pressure common-rail injection system. This system enables pilot injection before main injection. Thus the ignition delay time of the subsequent main injection is reduced considerably and combustion runs more gently, ultimately producing less combustion noise.

The injector is arranged centrally between the four valves and supports the target values for high efficiency and low emissions.

*Exhaust System and Turbocharger.* An *in-cylinder solution* and an oxidation catalyst reduce particulate and nitrogen oxide emissions considerably. A newly engineered piston combustion bowl likewise reduces pollutants.

Its compliance with the 2004 emission standards makes it possible to dispense with an external EGR line, which would add over 50 components to the engine's configuration and consequently make it more prone to failure. An expensive soot filter can be dispensed with for the same reasons.

A turbocharger with electronically controlled waste gate is integrated to further reduce emissions and reach the maximum power of 325 HP.

To spare the brake system when driving downhill, the Cummins 600 turbo diesel engine is equipped with an additional exhaust valve. It is closed as required and reduces the exhaust gas mass flow coming from the cylinder. This increases the in-cylinder pressure as a result of which the piston works against a stronger back pressure during the compression phase and crankshaft rotation is delayed.

### Modern V8 Gasoline Engine with Variable Valve Timing

This engine is descended from the modular V8 and V10 engine family (MOD for short), developed by Ford in 1991. The engines are variably designed in terms of their cylinder heads (two-, three- or four-valve cylinder heads) and their use (trucks and cars). In conjunction with the six speed automatic transmission, electronically controlled throttle, variable valve timing and other state of the art engine technologies, it was possible to develop an engine that satisfies the requirement for greater power while simultaneously reducing gasoline consumption.

This engine is currently used in the Ford Explorer and a slightly modified version is used in the Ford Mustang.

*Basic engine.* The basic engine is a 4.6 l unit. The eight cylinders arranged in a V shape have a bore of 90.2 mm and a stroke of 90 mm. This engine also has a cylinder angle of 90° often used for V8 engines (compensation for higher-order inertial forces and moments).

Depending on its vehicle use, the cylinder block is made of aluminum (Ford Mustang) or cast iron (Ford Explorer).

*Cylinder Head, Induction Pipe and Exhaust Manifold.* In contrast to the central crankcase, the cylinder head in the Ford Explorer as well as the Mustang is made of

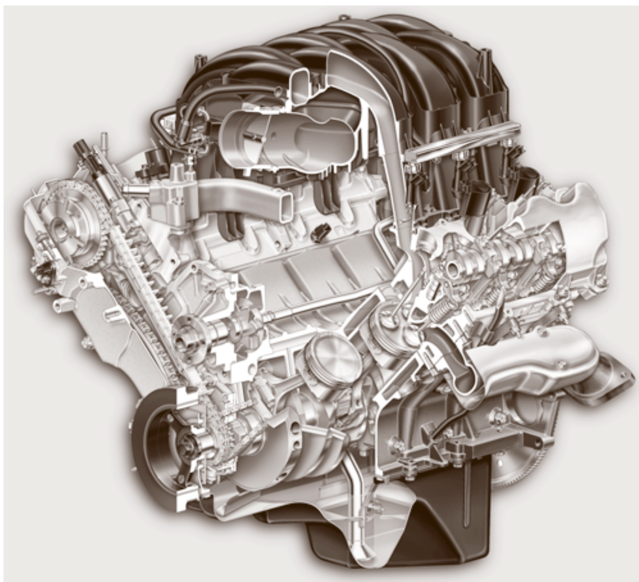


**Fig. 10.104** Ford 4.6l single overhead camshaft (SOHC) 90° V8-engine (illustration courtesy of the Ford Motor Company)

aluminum. The three valve cylinder head is lighter and smaller than the four valve variant. The new cylinder head enables a higher compression ratio of 9.8 : 1 when 87 octane fuel is used.

The large dual intake ports create a direct path to the intake valves for enhanced flow behavior at high rpm. At low rpm and engine loads, a processor-controlled charge motion control valve (CMCV) in the induction line closes shortly behind the injection nozzle. This considerably increases the flow velocity in the induction tract as well as the in-cylinder flow resulting in a more ignitable and faster combustible mixture. Consequently, in conjunction with the variable valve timing, an optimal charge motion characteristic can be achieved in the induction tract. As a result, fuel consumption drops by 10% compared to the predecessor model.

In addition, the flow conditions in the longitudinally optimized intake manifold could be noticeably improved and a discharge of combusted gases from the cylinder accelerated.

Due to their mass inertia, extremely light intake and outlet valves make high engine speeds possible and simultaneously reduce fuel consumption through their enhanced frictional properties.

To minimize valve gear noise, the cam covers were made of magnesium.

*Ignition System.* The three-valve technology allowed arranging the spark plugs centrally in the cylinder head. This results in three advantages:

- The central position to the cylinder produces a symmetrical flame with complete fuel combustion. Since the proportion of uncombusted fuel is negligible, the engine can generate more power while simultaneously reducing emissions (uncombusted hydrocarbons).
- The narrow and more-oblong design of the spark plugs make it possible to enlarge the valve diameters. This results in better engine performance and lower fuel consumption.
- A new powertrain control module (PCM) controls the ignitions more precisely, which ultimately manifests itself in higher efficiency.

*Variable Camshaft Timing (VCT).* After two- and four-valve engines in the modular engine family have been put in the widest variety of vehicles, a 24 valve cylinder head in the V8 variable valve timing is being implemented for the first time in 2005.

A single overhead camshaft per cylinder bank and low profile roller-finger followers with low friction activate the intake and exhaust valves. The powertrain control module electromagnetically changes the oil flow for the hydraulic cam timing mechanism, which enables the camshaft to rotate opposite the drive sprockets. The mechanism can switch between fully advanced and fully retarded timing in only a few milliseconds.

VCT achieves an angular camshaft control of 50° CA. The *dual-equal* camshaft timing developed by Ford changes the intake and exhaust valve timing simultaneously. This system provides decisive advantages over fixed timing in the engine's complete speed range. Short seat timing at low speeds causes the cylinder pressure to drop less strongly as a result of which a high torque is generated. The slight valve overlap also reduces emissions. The seat timing increases at high speeds. The greater charge mass in the cylinder also increases engine performance.

This synchronous control of the timing enables constructing the cylinder head less complexly and with less weight than fully variable systems in which the intake valve is controlled separately from the exhaust valve.

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